THE

DESIGNING OF DRAW-SPANS.

IN TWO PARTS.

I. PLATE-GIRDER DRAWS.

II. RIVETED-TRUSS AND PIN-CONNECTED LONG-SPAN DRAWS.

BY

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PREFACE.

THE author has desired to collect in this volume all the material necessary for the complete designing of the common types of draw-spans, and to arrange it in such form as to be of the greatest service in practical use. Several special types, such as the rolling-lift and the vertical-lift spans in use at Chicago and elsewhere, and the folding draw as used in Boston, are considered and their special features illustrated.

The points which have received the most attention are those which the author has failed to find very fully discussed in works on draw-spans, and which are, many of them, the things which cause the young engineer or the one not familiar with draw-spans the most trouble when the practical design of a draw-span is undertaken.

The author wishes to express his thanks to the *Engineering News* and the *Engineering Record* for numerous favors and the use of cuts and material generously permitted by them, and to many engineers whose bridges are described in this work, for their courteous response to requests for information.

CONTENTS.

DESIGNING OF DRAW-SPANS.

PART FIRST.

PLATE-GIRDER DRAW-SPANS.

THE following pages aim to give a clear and simple explanation of the methods used in the determination of the stresses, sections required, and of the deflections produced by the various conditions of loading assumed. The machinery necessary for operating the draw is also considered, and the designing of wedging machinery for raising the ends, latching devices for preserving perfect alignment when the draw is closed, methods of raising the rails for clearance when the draw is opened, and the designing of gears, shafting, and bearings are considered in detail. Each point as taken up is illustrated by examples, as fully as necessary to make the applications clear. The aim has been to use the simplest methods, rules, and tables that will give the desired results. Where formulæ derived from the higher mathematics have been used, full and complete explanations of how they are used and applied are given.* It is believed the work may be

* The reader is referred to the works of Professor Releaux and Unwin, from which notes have been taken. The author is also indebted for valuable information to Professor Malverd A. Howe of Rose Polytechnic Institute; the Edge Moor Bridge Works, and to the Pencoyd Bridge Works.

readily followed and understood by those not having a full knowledge of the higher mathematics, and that it will prove of value to any one wishing a practical knowledge ef draw-spans and their machinery.

Plate-girder Draws.

For spans up to about one hundred and fifty feet the deck plate girder makes the most satisfactory bridge, and is the type in most general use. The conditions under which the draw-span works are much more severe than with fixed spans, and the bridge should be correspondingly heavy and rigid. Through plate-girder or lattice spans are unsatisfactory for draw-spans, owing to the small depth usually available below the floor for the introduction of diagonal bracing necessary to resist the twisting force produced in turning the draw, and especially in suddenly stopping or starting. This force is well illustrated by taking a piece of artist's rubber in the fingers and twisting. The rubber may be turned through a considerable angle and still a cross-section at any point will be a perfect rectangle as at first. This shows that any bracing introduced to resist this twisting action must run diagonally as in Fig. 1 and 1^A. Brace-frames at right angles to the girders do little good to resist such a force, and the same is true of bracing in the planes of the chords.

An eighty-five-foot deck plate girder (Fig. 1) will be used as an example to illustrate the methods pursued with girder draws in general. There are four conditions to be considered. 1st. The span swinging or in position to open, the end wedges being drawn and all the dead loads being carried by the centre, no live load acting. 2d. The draw closed and each arm considered as an independent span for live load; the dead load not being considered for the present. 3d. The bridge

considered as two continuous spans for the live load; and 4th, considered as two continuous spans for the dead load onlv

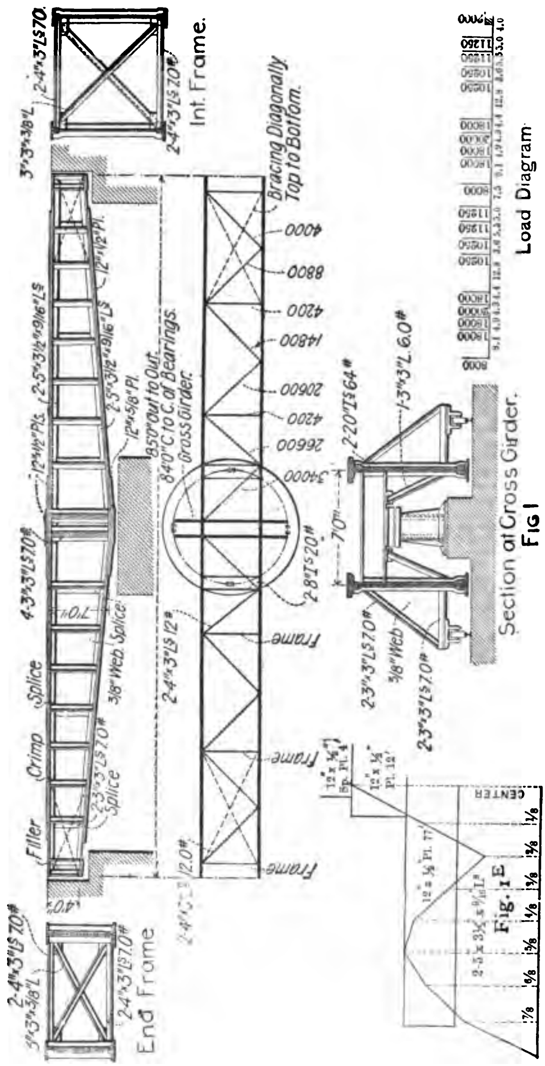


Fig I

Cases 1 and 2 might occur at the same time; also 1 and 3 or 3 and 4. The one of these combinations giving the

greatest strains is to be used in determining the sections required.

If the end wedges are just driven to a bearing but not hard enough to *raise* the ends, the dead load would still be carried by the centre, and the span is still *swinging* so far as the *dead* load is concerned. If both arms were now loaded *equally*, the bridge is then a continuous girder of two spans so far as the *live* load is concerned. This is not true, however, if a live load comes on one arm only, unless the other arm be held down so that it does not raise up off the end support as the dead load moves over the first arm. Instead of holding the unloaded arm down, it may be raised so high by the end wedges that the deflection produced in the loaded arm will not be sufficient to raise the unloaded end off the support. Unless one or the other of these plans is followed there will be what is called 'hammer' in the draw. That is, as the load comes on one end and moves over the bridge, first one end and then the other will rise off the supports and drop back again to a bearing. This movement is very noticeable in some draws, and especially so where the rails are cut just at the clearance line and a small space left between the ends. To make sure the rails will clear as the draw turns, this space may needs be three-eighths or one-half inch. This method, or lack of method, of providing for the continuity of the rails is now almost entirely superseded by devices which do not require this clearance. Some of the methods used will be described later. The amount it is necessary to raise the ends by means of the wedges or some similar device will be explained under the deflection of draws.

To determine the strains produced by the dead load swinging, we will assume the weight of the floor (including ties, guards, rails, bolts, etc.) to be 400 lbs. per linear foot, and the weight of the span itself to be 650 lbs. per linear foot. $400 + 650 = 1050$ lbs. $= 525$ lbs. for each girder. Only one arm need be considered if the two arms are equal. If the

two arms are not equal, the shorter one is counterweighted until they balance, but the strains would have to be considered separately. The moments may be determined by assuming the dead load as concentrated at several points; thus for the moment over the pier we may assume the load on one arm as concentrated at its centre of gravity, which is at the centre of the arm (see Fig. 2).

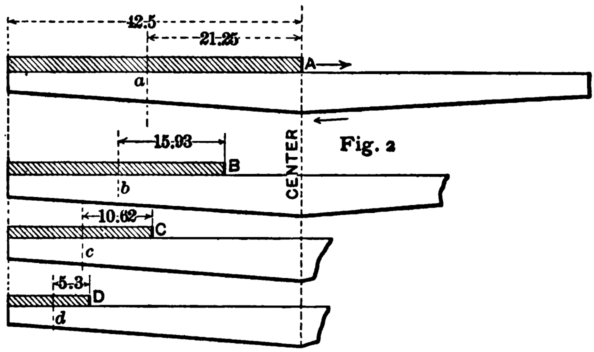


Fig. 2

Arms for Dead-load Moments.

Taking moments at A, we have the dead load of one arm, $525 \times 42.5 = 22,310$. Assuming this as concentrated at a, the moment is $22,310 \times 21.25 = 474,100$ ft.-lbs. This moment is balanced by forces represented by the arrows and acting in the flanges of the girder. One force is tension and the other compression. The depth of the girder at the centre is 7 feet and the moment $474,100 \div 7 = 67,750$, which is the tension in the upper flange and the compression in the lower.

The depth assumed (7′) should be the depth between the centres of gravity of the flanges. For the moment at B we have the load 525×31.86 (the distance from the end to B) $= 16,730$. This multiplied by the distance of the centre of this load from B, 15.93 feet, $= 266,500$. Dividing by the depth at this point, 6.25 feet, we have $266,500 \div 6.25 = 42,800$. At C the moment $= 525 \times 21.25 \times 10.62 = 118,470$. At D the moment $= 525 \times 10.62 \times 5.31 = 29,600$. It is not

necessary to find the chord stresses at each point now. The moments may be combined with others for live load, and the areas required for both found at one operation. The moment at any point for dead load may also be found by means of a parabola drawn as follows (Fig. 3). Lay off the horizontal

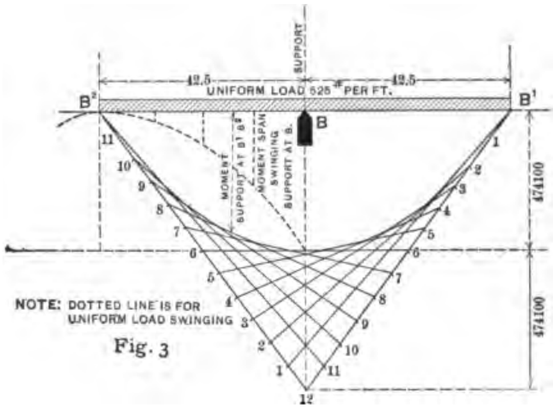


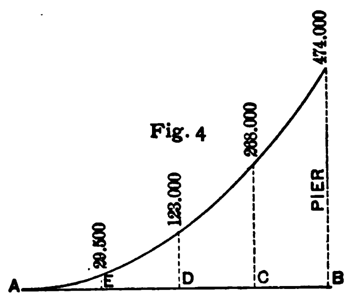
Fig. 3

METHOD OF DRAWING PARABOLA

line BB' equal to twice the length of one arm of the draw. From the centre of this line draw a vertical line equal to twice the moment at the pier * (in this case 474,000). Any convenient scale may be used, and the same scale need not necessarily be used for the horizontal and the vertical lines. Draw the inclined lines B 12 and B' 12, and divide each of them into any number of equal parts (12 in the figure). Connect the points 1–1, 2–2, 3–3, etc., and the lines so drawn will be tangents to the required curve, which is now readily drawn. Only one half the curve is used, as shown by the figure. The curve being drawn, the moment at any point is

* To find the centre of gravity of any number of loads from any point (as one of the end loads), multiply each load by its distance from the point, add the results, and divide by the sum of all the loads. The result will be the distance of the centre of gravity from the point assumed. Note that if there is a load at the point from which we start, this load must be included in getting the sum of all the loads.

found simply by scaling the ordinate between the line $B'B$ and the dotted curve. Having thus shown two methods for determining the dead-load moments with the draw swinging,



Curve of Moments, Dead Load Swinging. 525 lbs. per lin. ft.

we will now consider the case of the draw closed and each arm acting as a single span for *live* load.*

For the live-load moments, each arm acting as a single span, we should so arrange the loads as to get as many loads as possible on the span, and the heavier ones as near the centre as may be. Placing the loads as in Fig. 5, we find the centre of gravity to be 18.7 feet from wheel No. 1, and the wheels are shifted if need be until the centre of the span is half-way between the centre of gravity and load No. 4. We now lay off the load line AB, Fig. 5^A, assume a distance $HO = 100{,}000$ on a horizontal line drawn from any point in AB, and draw the lines AO, BO, etc., connecting the points found by laying off the loads on AB with the point O. This figure (5^A) is called the force polygon. Next, starting from A' (any point in a vertical line through A) draw the line $A'a'$ parallel to AO in the force polygon, and from a' draw the line $a'b'$ parallel to $5-O$, from b' the line $b'c'$ parallel to $4-O$, and so on until the last line $f'B'$ is drawn parallel to BO.

* In drawing the parabola it will be noticed that the moment over the pier must first be figured. This moment for the load uniformly distributed is $\frac{1}{2}wL$, L being the length of the arm, and w the dead load of one arm. (See first method of finding the dead-load moments.)

The line $A'a'b'c'-f'B'$ meeting the vertical lines through A and B at A and B' is called the equilibrium polygon. If the line OR be drawn in the force polygon parallel to $A'B'$ of the equilibrium polygon, it will divide the load line AB into

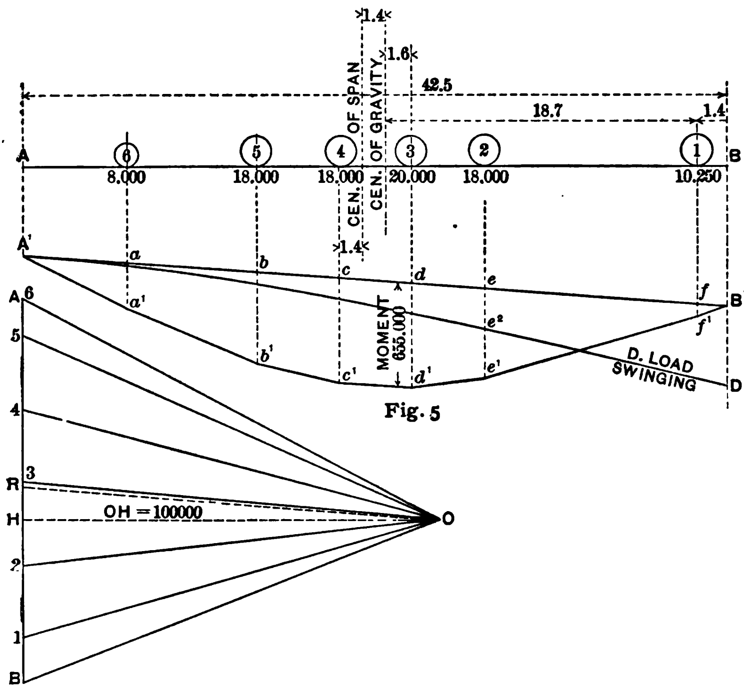


Fig. 5

Diagram for One Arm as Single Span.
Moment at any point as $CC' = CC' \times HO = CC' \times 100$.

the two parts AR and RB which represent the reactions at A and B. Having the equilibrium polygon drawn, the moment at any point is found by multiplying the ordinate between the closing line $A'B'$ and the line $A'a'b'c'$, etc., by the distance OH in the force polygon. HO being 100,000, the moment at b, for example, will be bb' multiplied by 100,000. The distance HO is made 100,000 for convenience. It should be made of such length as will give a good

depth to the equilibrium curve, so that the ordinates may be accurately scaled. The distance HO must be measured to the same scale as the load line AB was laid off, and the ordinates in the equilibrium polygon must be measured to the scale used in laying off the half-length of the span (see Fig. 5). It is not necessary that the two figures be drawn to the same scale. The moments at as many points as necessary can now be determined. These moments are given in column 4 of the table of strains. In Fig. 5 the curved line $A'D$ and the line $A'B'$ give the dead-load moments with the span swinging, $A'D$ being a parabola and the ordinate $B'D$ being the moment at the centre support divided by the distance HO (= 100,000). The signs of the moments are determined as follows: The loads acting to the left of the centre support tend to revolve the span downward in a direction opposite to the movement of the hands of a clock. These moments are called minus (—). Considering the same arm as a single span, the reaction at the left support tends to revolve the span upward or in the direction of clock motion. These moments are called plus (+). It is immaterial which are called plus, provided all moments tending to produce rotation in the same direction are given the same sign. The total moment then at any point, as e', under the two conditions, dead load swinging and live load discontinuous, on one arm, would be the ordinate $ee' - ee^2 = e^2e'$ multiplied by the pole distance HO. It might be found that slightly greater moments would be obtained by placing the loads so that the centre of the span would be between the centre of gravity and load number 3, instead of between the centre of gravity and load number 4 (see Fig. 6). Both positions should be tried. Having shown how to determine the moments for the span swinging, and for the condition of one arm acting as a single span supported at the ends, with live load only acting, we will now consider the span as a continuous girder under the action of both dead and live load. It

will be noted that in the case of dead load swinging only one arm was considered. This is sometimes confusing and the question is asked, 'Why can one arm be neglected? They must surely both produce strains over the centre.' It is the old problem of two men pulling at the ends of a rope; each man pulls one hundred pounds, but the strain on the rope is not two hundred pounds. One man cannot pull one hundred pounds unless there is a resistance of this amount opposing his pull. It makes no difference whether the resistance is given by a man or by a post at the other end of the line. In the same way an arm of the draw when open is balanced by the other arm. And the moment at the centre is the moment produced by one arm. When the span rests on three or more supports or the loads are not balanced we can no longer consider one arm only.

If a load is placed at any point on the span, a greater proportion of this load will be carried to the centre support than would be the case if the arm on which the load is placed were considered as a single span resting on two supports. Just how much more of the load is carried to the centre is given by the diagram Fig. 9. The figures at the bottom under the line 'values of k' are the distances from the left-hand support to the loaded point, in terms of the length, and the figures in the line marked 'values of D_1' give the per cent of the load going to the left-hand support. Suppose there is a load at three tenths of the length of the arm from the left support. From the figure 0.3 in line k_1 we move up until this line intersects the curve marked 'S_1 loads in first arm'; from the point where the line through 0.3 intersects this curve we go over to the left until we reach the line D_1, which is at 0.63. 63 per cent of the load then goes to the left support. If we wish for the bending moment at this point, we move up the line through 0.3 in k_1 until we meet the curve marked 'M_2 loads in first or second arm.' We intersect

this curve on the horizontal line 0.685,* and so for any other point in the span. We will now place the engine-loads on the span in two or three positions and see which position will

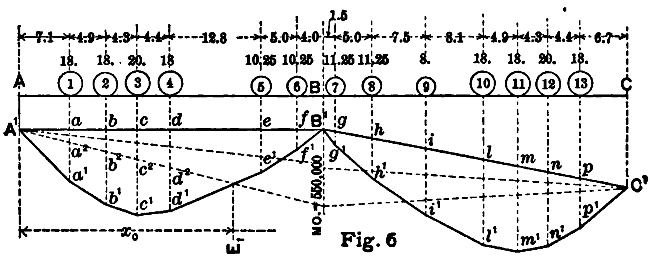


Fig. 6

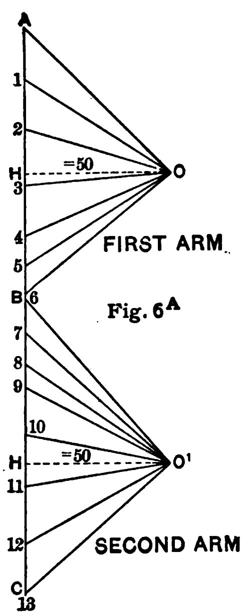


Fig. 6A

FIRST ARM.

	k.	*c.*	*CPL.*
7.1 ÷ 42.5	= .167	.0405	30,980
12.	= .282	.0645	49,340
16.3	= .384	.0819	69,610
20.7	= .487	.0927	70,900
33.5	= .788	.0745	32,450
38.5	= .906	.0400	17,420
			270,700
Moment = *CPL*			279,385
			550,085

SECOND ARM.

	k.	*c.*	*CPL.*
6.7 ÷ 42.5	= .0156	.0380	29,608
11.1	= .261	.0605	51,450
15.4	= .362	.0785	60,030
20.3	= .477	.0920	70,380
28.4	= .668	.0925	31,350
35.9	= .845	.0600	28,680
40.9	= .962	.0165	7,887
			279,385

Diagram for Two Spans Continuous. Scales, 20 and 50.

* 0.685 is the value of *C* in formula $M = CPL$ = moment at any point.

give us the greatest moment over the pier. Arranging the loads as in Fig. 6, we first find the values of k; thus for loads

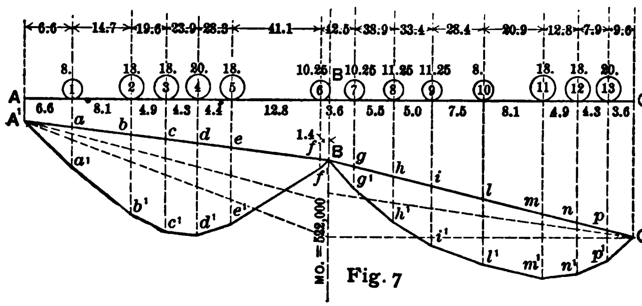


Fig. 7

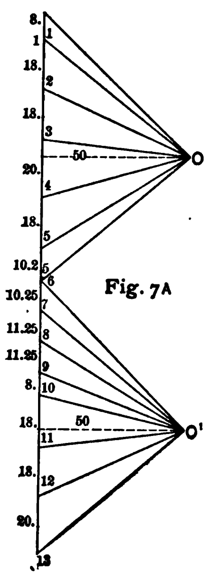


Fig. 7A

FIRST ARM.

k.	c.	CPL.
6.6 ÷ 42.5 = .155	.0380	12,929
14.7 = .346	.0756	57,815
19.6 = .461	.0910	69,605
23.9 = .562	.0963	81,850
28.3 = .666	.0927	70,905
41.1 = .967	.0150	6,525
		299,629
Moment = CPL		233,331
		532,960

SECOND ARM.

k.	c.	CPL.
3.6 ÷ 42.5 = .084	.0215	18,274
7.9 = .186	.0450	34,422
12.8 = .301	.0685	52,395
20.9 = .492	.0933	31,720
28.4 = .668	.0927	44,760
33.4 = .786	.0750	35,862
38.9 = .915	.0365	15,898
		233,331

Diagram for Two Spans Continuous. Scales, 20 and 50.

1, 2, 3, 4, 5, and 6 we divide the distances from the left by the half-span 42.5′, and for loads 7, 8, 9, 10, 11, 12, and 13 we divide the distances of the loads from the right-hand abut-

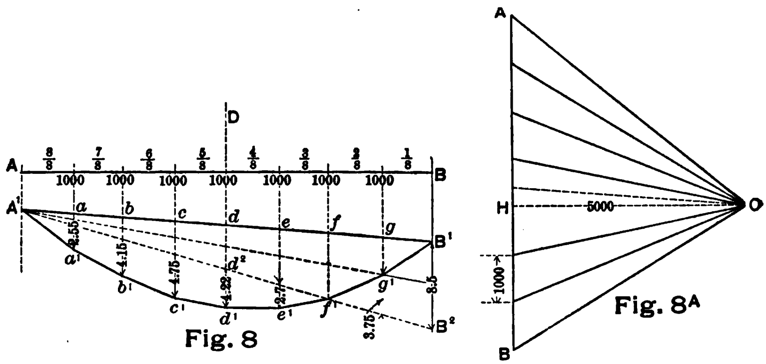


Fig. 8

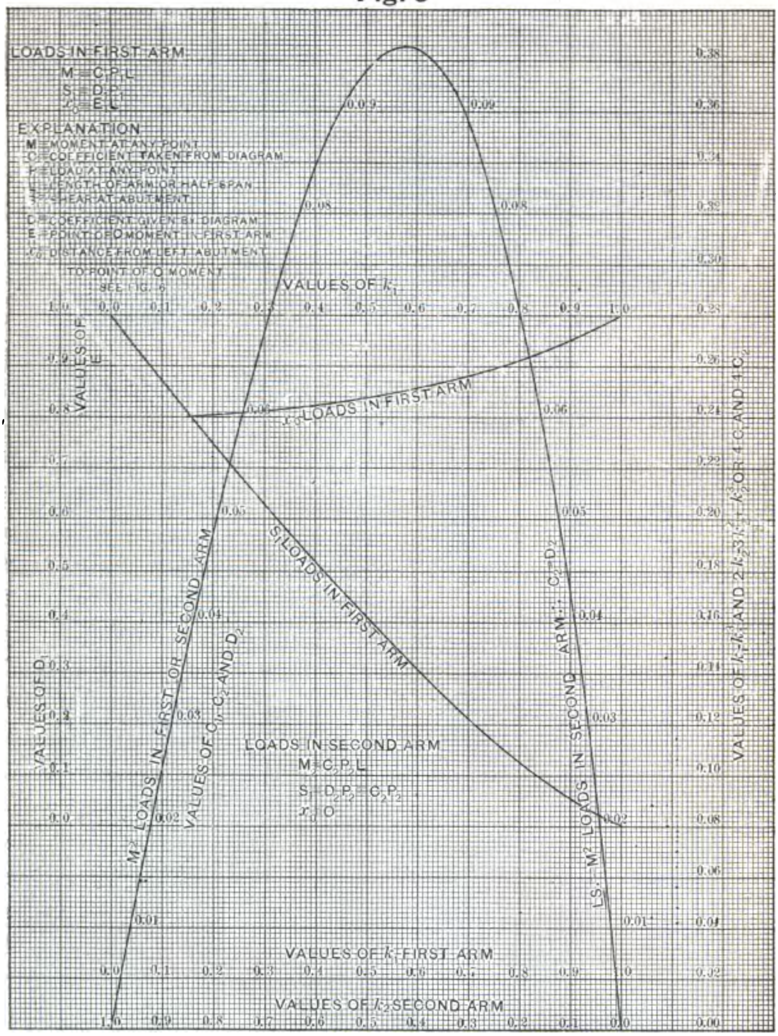
Fig. 8A

Diagram for Uniform Load Continuous. 1000 lbs. at each eighth point assumed load in diagram.

ment by the half-span 42.5. The values are given in the table: .167, .282, etc., for first arm and .0156, .261, etc., for the second arm. From diagram Fig. 9 we now find the values of C corresponding. The vertical through $k = 167$ meets the curve of moments on the horizontal line .0405, and for $k = .788$ on the line .0745. The values of k for the second arm are given from the right abutment, so we find C exactly as in the first arm. If the distances had been given from the centre pier, we could have found C in the same manner, only using the line marked k^2 in the diagram instead of line k'; for example, if a load is .8 the length of the half-span from the right abutment, it is .2 the half-arm from the centre pier. $0.1k'$ is over $k^2 = 0.9$. It is perhaps a little simpler to use the line k' all the time, and give the distances of the loads from the abutments in each case. All values of C have the same sign. Multiplying each value of C by the *load at that point*, and by the length of the half-span, gives us the moment over the centre pier for that load. $CPL =$ moment over pier for load P at any point. The values of

these moments for each of the wheel-loads with the engine placed in the two positions given in Figs. 6 and 7 are given

Fig. 9



in the tables under the figures. Two or three trials will show how the engine should be placed to give the greatest

moments. By referring to the diagram Fig. 9 it will be seen that C is greatest for loads near the centre of each arm, and a little nearer the centre pier than the abutments. The heavier wheels should then be placed as near these positions as possible to give the maximum moments. Adding together the moments produced by all the loads, we have the total moment. In the two cases given these total moments are 550,085 and 522,960. It is possible that the uniform train load might give a greater moment at the pier than the engines, and this moment should be found.

Before considering the uniform load we will take one more example of moment from concentrated load to make the method just described perfectly plain.

Suppose we take wheel No. 11 in Fig. 7. The distance of this wheel from the right abutment is 12.8. $k = 12.8 \div 42.5 = .301$. C for $k = .301$ is .0685, and CPL, the moment, $=$ 52,395. $P = 18$ and $L = 425$.

Considering now the case of uniform load, span continuous, the Reading loading diagram gives 4000 lbs. per linear foot, or 2000 lbs. on one girder. $2000 \times 42.5 = 85,000 =$ the load on one arm. The formula for the moment at centre support with uniform load is $\frac{1}{8}wl^2$ $w =$ the load per foot, and $l =$ the length of one arm of the span In this case $w =$ 2000, $l = 42.5$, $wl = 85,000$, $\frac{1}{8}wl^2 = 451.562$. This is considerably less than the moment from the wheel-loads, which was 550,085 for one position of the loads. It will be noticed that the moment over the pier, $\frac{1}{8}wl^2$, is just the same as the moment at the centre of a single span of length equal to one arm of the draw and covered with the same uniform load; and is also just one fourth as much as it would be over the centre support were the draw swinging and covered with the same load. Note that in moment $\frac{1}{8}wl^2$, wl is load on one arm. A convenient method of finding these moments for uniform load is to assume a load one pound or one thousand pounds per foot, find the moments for this loading, and then

multiply the results by the ratio of the actual loads to the one assumed. To make us a little more familiar with the force and equilibrium polygons, we will divide each arm into eight parts and assume a load of 1000 lbs. at each of these points and one half load at the ends. The loads at the ends, coming directly over the supports, may be neglected in the computation. We lay off then on the vertical line AB, Fig. 8^A, seven spaces representing 1000 lbs. each. Any scale may be used, say one-half inch equals 1000 lbs. Next assume the point O distant from AB 5000 to the same scale. Note that the point O may be anywhere in a vertical line which is distant 5000 from the vertical line AB, and also remember that we assumed the distance 5000; any convenient distance may be used. We next connect the point O with each of the points laid off on AB. Now going to Fig. 8, at any point on a vertical through A we draw the line $A'a'$ parallel to AO in Fig. 8^A, and from a' the line $a'b'$ parallel to the next line in the force polygon, and so on until finally $g'B$ is drawn parallel to BO in the force polygon. Now connect A' and B' with a straight line. From B' in Fig. 8 scale off the distance B'–B^2 equal to the moment at the centre support divided by the distance $HO = 5000$ in Fig. 8^A. The distance $B'B^2$ must be laid off to the same scale as Fig. 8 is drawn to. The moment at the centre is of course found for the same loading (1000 lbs. at each eighth point $= \frac{1}{8}wl^2$). By the diagram Fig. 9 the values of C for $k = \frac{1}{8}, \frac{2}{8}, \frac{3}{8}, \frac{4}{8}, \frac{5}{8}, \frac{6}{8}$, and $\frac{7}{8}$ are:

$K = \frac{1}{8} = .125$	$C = .0308$;	$P = 1000$,	$L = 42.5$;	CPL	$= 1309.00$
$K = \frac{2}{8} = .250$	$C = .0586$;	"	"	"	$= 2490\ 50$
$K = \frac{3}{8} = .375$	$C = .0806$;	"	"	"	$= 3425.50$
$K = \frac{4}{8} = .500$	$C = .0938$;	"	"	"	$= 3986.50$
$K = \frac{5}{8} = .625$	$C = .0952$;	"	"	"	$= 4046.00$
$K = \frac{6}{8} = .750$	$C = .0820$;	"	"	"	$= 3485.00$
$K = \frac{7}{8} = .875$	$C = .0513$;	"	"	"	$= 2180.25$
	$.4923$				20922.25

The moment $\frac{1}{8}wl$ for the same load uniformly distributed (8000 lbs. on each arm) is 42,500. The difference by the two

methods is 655.75 or 1½ per cent, which shows that the method is practically correct, and it is merely a question of reading the diagram correctly to obtain accurate results. Making a table of the moments (see p. 18), we have first the column of moments for dead load swinging, the moments being found by methods shown in Fig. 2 or 4. These moments are 474,000, 350,000, etc. Next we make the column of moments for dead load continuous, as shown by Fig. 8, remembering that the moment at any point is equal to the moment for the same load, considering the arm as a single span supported at the ends, less the negative moment at this point, and that this negative moment is represented by the ordinate between the lines $A'B'$ and $A'B^2$ multiplied by the pole distance HO; the ordinate B^1B^2 being the moment over pier divided by the pole distance HO. Thus the moment at D equals ordinate dd' minus ordinate dd^2 (Fig. 8) multiplied by HO ($HO = 5000$).

Having the moments tabulated, we now see which combinations will give the largest totals. The dead load swinging and live load continuous, case A, give the largest moment over the centre support, 1,024,000. The same combination also gives the greatest moment at the ⅞ point. At the quarter point the dead load swinging and case A live continuous give a minus moment of 318,000, and live load discontinuous with dead load swinging give a plus moment of 187,000, and so at each of the points ⅜, ⅛, etc., we obtain the results given in column 8. Dividing these results by the depth of girder (centre to centre of gravity of flanges), we obtain the results given in column 10. Dividing these results by the unit stresses as allowed by the specifications (in this case 8000 lbs.), we have the areas required (column 12).* In Fig. 1^E the areas required at the several points are laid off to scale, and the lengths of the cover-plates required readily determined.

* Where the flange-areas are determined for tension, the areas after deducting rivet-holes must be used.

TABLE OF STRAINS.

Dist. from Pier.	Moment.				Live Load Continuous.		Total.	Depth.	Flange-strain.	Unit Stess.	Area.
	Dead Load Swinging.	Dead Load Continuous.	Live Load Discontinuous.	Live Load Uniform, 2000 lbs.	Fig. 6. Case A.	Fig. 7. Case B.					
1	2	3	4	5	6	7	8	9	10	11	12
Pier	− 474,000	− 118,500		− 451,600	− 550,000	− 522,000	− 1,024,000	7′.	146,280	÷ 9,000	= 18.29
1/8	− 350,000	− 50,900	+ 255,000	− 194,000	− 270,000	− 250,000	− 620,000	6′.63	93,500	"	= 11.69
							− 318,000		47,000	"	= 5.9
2/8	− 268,000	− 00	+ 455,000	− 00	− 50,000	+ 45,000	+ 187,000	6′.25	29,920	"	= 3.75
							− 185,000				
3/8	− 185,000	+ 38,060	+ 600,000	+ 145,000	+ 156,000	+ 273,500	+ 415,000	5′.88	68,880	"	= 8.61
							− 123,000				
4/8	− 123,000	+ 61,510	+ 655,000	+ 226,000	+ 317,000	+ 395,000	+ 532,000	5′.50	96,720	"	= 12.1
							− 65,000				
5/8	− 65,000	+ 66,700	+ 600,000	+ 254,000	+ 417,000	+ 400,000	+ 535,000	5′.13	104,300	"	= 13.04
							− 29,500				
6/8	− 29,500	+ 59,500	+ 455,000	+ 227,000	+ 382,000	+ 300,000	+ 441,500	4′.75	92,900	"	= 11.61
							− 10,000				
7/8	− 10,000	+ 37,300	+ 255,000	+ 142,000	+ 225,000	+ 200,000	+ 262,300	4′.38	60,000	"	= 7.50

Combinations: Col. 2 with 4–6 or 7; col. 3 with 6 or 7.

The plates should extend about two feet beyond the points so determined.

The web is not considered as taking any flange-stress, and the area in top flange is made up by two $5'' \times 3\frac{1}{2}'' \times \frac{9}{16}''$ angles and two $12 \times \frac{1}{2}$ plates. One of the plates will be too long to get in one length, and a splice-plate is added to make up the section at the splice. In the bottom flange two $\frac{5}{8}''$ plates are used.

WEB-STRAINS.

We will next consider the shearing stresses in the web. The greatest shear at the abutments will be obtained by considering one arm as a single span for live load and dead load swinging, no dead reaction at abutment, as the condition of dead load *continuous* and live load discontinuous cannot occur. See combination of strains made. From a table of 'shears and bending moments' for this engine we have the end shear for a span 42.5 = 72,650 lbs. That is, 72,650 lbs. is the *upward* force exerted by the support at the abutment. Say the specifications allow 6000 lbs. per square inch shearing on webs; then $72{,}650 \div 6000 = 12.1$ sq. in. required; $48 \times \frac{3}{8}$-inch web plate gives 18 sq. in. At the quarter point the upward shear is 46,500 lbs. From this is to be taken the dead load between the abutment and this point. This load equals $525 \times 10.62 = 5600$ lbs. $42{,}500 - 5600 = 36{,}900$ lbs. Note that in finding the greatest live-load shears the heavy wheel at the front of the engine is placed at the point where the shear is required, and that there is no live load on the span between the abutment and the point whose shear is being determined. At the centre of the arm the live shear is 22,700 upward, and the dead-load shear downward is 11,200. $22{,}700 - 11{,}200 = 10{,}500$. The greatest shear at the pier will be with dead load swinging (all dead load carried to the

pier) and with live either continuous or discontinuous. For discontinuous live load we have the same maximum shear at the pier as at the abutment, the engine simply headed the

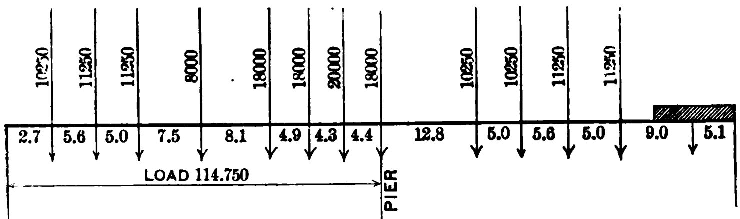


Fig. 10

	Span.	k_1	D_1	Load.	S_1
2.7	425	0.06	0.93	10,250	9,740
8.3	"	0.19	0.76	11,250	7.780
13.3	"	0.31	0.62	11,250	6,975
20.8	"	0.49	0.41	8,000	3,280
28.9	"	0.68	0.23	18,000	4,140
33.8	"	0.79	0.14	18,000	2,520
38.1	"	0.89	0.06	20,000	0,120
42.5	"	1.00	0.00	18,000	0,000
					34,355
			D_2		
12.8	425	0.30	0.089	10,250	912
17.8	"	0.42	0.096	10,250	984
23.4	"	0.54	0.091	11,250	1,024
28.4	"	0.68	0.071	11,250	782
37.4	"	0.88	0.024	20,000	481
42.5	"	1.00	0.0		000
					4,183

$$114,750 - (34,355 - 4183) = 84,580 \pm .$$

Shear at Centre, Girder Continuous.

other way. We have then the upward shear live $= 72,650 +$ the dead weight of one arm $= 22,300$. $72,650 + 22,300 = 94,950$.

Considering now the case of live load continuous: it is clear that a load in any position (as the centre) on one arm

tends, by causing this arm to deflect, to raise the other arm off its abutment or end support. This support then has less to do or the shear is reduced at this point by the load on the other arm; it follows therefore that, as all the load on the span must be carried by the abutments and the pier, if some load is taken from the abutment it must be added to the load on the pier. A greater proportion of the load is carried by the

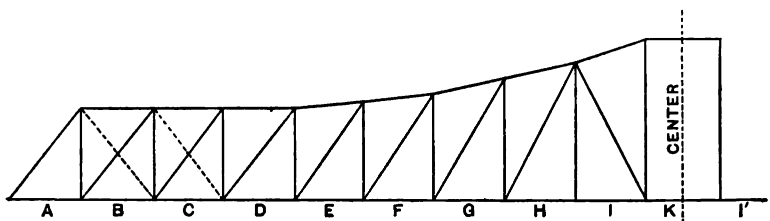


Fig. 1-

centre pier considering the two arms as continuous than by considering them as independent spans. And in determining the shear at any point the loads on both arms must be considered. By means of diagram Fig. 9 the reactions caused by loads at any point in either arm are readily determined. Arranging the loads as in Fig. 10, and finding the values of k_1, k_2, and D_1, D_2, we get for the shear just to the left of the

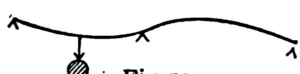


Fig. 12

pier 84,580; this added to the dead-load shear gives a total of $84,580 + 22,300 = 106,880$. The area of $84 \times \frac{3}{8}$ web $=$ 31.5 sq. in., against $106,880 \div 6000 = 17.8$ required. Stiffeners should be at intervals of about the depth of the web apart, with 6 ft. as a maximum.

Lateral Bracing.

The laterals should be figured for a wind-load of, say, 600 lbs. per lineal foot, the point being to get sections heavy enough to render the span stiff laterally. Cross-frames should be used at intervals of ten to fifteen feet. Note that lateral bracing should be figured to carry strains to the centre, and that this force, equalling at least 300 lbs. per foot = 25,500 lbs. for both arms, should be considered in designing centre pivot and anchorage.

Cross-girders.

When the draw is closed and ready for the passage of trains the girders are supported at the centre by wedges, so the cross-girders carry only the dead weight of the span. This amounts to 44,600 at each side; as there are two cross-girders, the moment on each is $22,300 \times 42$ in. $= 936,600$ in.-lbs. Using 20-in. 64-lb. beams, with a moment of resistance of 114, gives a fibre-stress of 8200, allowing an ample margin.

Centre-post.

The load on the centre-post is about 90,000 lbs. The base of the post should be large enough to distribute this well over the masonry and to give the post stability. There should be anchor-bolts built into the masonry, and their area should be sufficient to resist the shear from wind-forces, assuming for this purpose a wind-pressure of 300 lbs. per lineal foot of bridge, and neglecting the friction of the base-plate on the masonry. This gives a force of $300 \times 85 = 25,500$ lbs. Four $1\frac{1}{4}$-in. bolts at 7300 lbs. each would be ample. A wrought-steel post is preferable to one of cast iron, as it is much less liable to break if the bearing on masonry becomes unequally distributed. The post should be made high enough

to throw the point of suspension into the upper half of the web; the girders will then hang better and turn more easily, as there will be less weight thrown on the trailing-wheels.

DEFLECTION.

Deflection Formulæ.

NOTE.—These formulæ are applicable to spans of any length if the proportions are approximately as given below

$$I = \frac{\left(\frac{1}{2}h\right)^2\left(\frac{1}{2}l + x\right)}{12}.$$

$$D \text{ for uniform load} = \frac{4.704\,WL^2}{Eh^2} \quad \ldots \ldots \quad (1)$$

$$D \text{ for load at end} = \frac{13.18\,PL^2}{Eh^2} \quad \ldots \ldots \quad (2)$$

$$I = \frac{1}{6.8}h^3. \qquad h = \frac{4}{7}h_1 + \frac{3}{7}h_1 \cdot \frac{x}{l}$$

$$D \text{ for uniform load} = \frac{1.166\,WL^3}{Eh_1^3} \quad \ldots \ldots \quad (3)$$

$$D \text{ for load at end} = \frac{3.377\,PL^3}{Eh_1^3} \quad \ldots \ldots \quad (4)$$

$$I = \frac{1}{6}h^3. \qquad h = \frac{1}{3}h_1 + \frac{2}{3}h_1 \cdot \frac{x}{l} = \frac{2}{3}\frac{h_1}{l}\left(\frac{l}{2} + x\right).$$

$$D \text{ for uniform load} = \frac{1.315\,WL^3}{Eh_1^3} \quad \ldots \ldots \quad (5)$$

$$D \text{ for load at end} = \frac{4.248\,PL^3}{Eh_1^3} \quad \ldots \ldots \quad (6)$$

D = deflection;

h_1 = height at centre; h = height for any distance x;

L = length in inches;

x = distance from left end in inches;

P = load at end; W = totol load uniformly distributed.

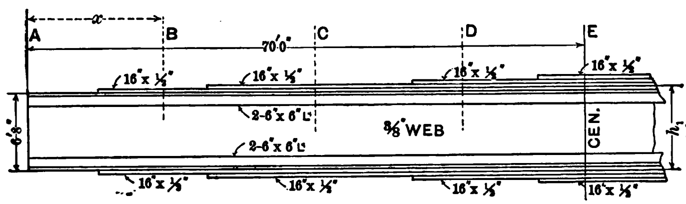


Fig. 13.

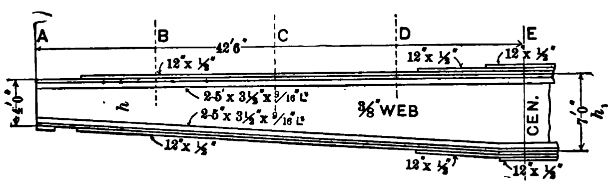


Fig. 14.

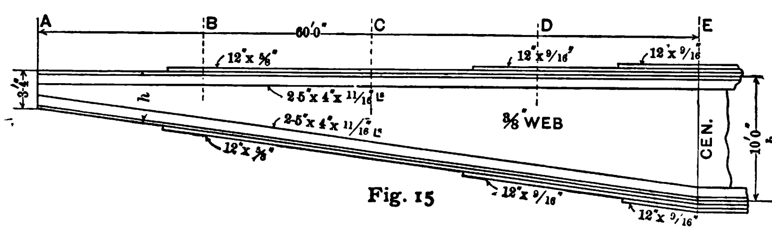


Fig. 15

The amount the girders will deflect under the various loads depends upon the length, the depth, and the arrangement of the material in the girders. If the flanges are parallel and their area of cross-section remain the same or nearly so

throughout their length, the formula for deflection for constant section may be used; thus for uniform load

$$D = \frac{Wl^3}{8EI}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$D =$ deflection, $W =$ the load on the girder, $l =$ the length in inches, $E = 29,000,000$, and $I =$ the moment of inertia. For a load at the end

$$D = \frac{Pl^3}{3EI}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

If the flanges are parallel, but the cover-plates are of several lengths and the girder have about the proportions shown in Fig. 13, the deflection for uniform load will be

$$D = \frac{4.704\,Wl^3}{Eh_1^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$W =$ total load on arm, and $h =$ depth of girder back to back of flange-angles. For load at end

$$D = \frac{13.18Pl^3}{Eh_1^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Number 1 is equal to

$$D = \frac{Wl^3}{195,500,000I}, \quad . \quad . \quad . \quad . \quad . \quad (1a)$$

and number 2 may be written

$$D = \frac{Pl^3}{68,900,000I}. \quad . \quad . \quad . \quad . \quad . \quad (2a)$$

$W =$ total load on one arm in each case, and $I =$ the moment of inertia at the centre support. For girders having approximately the proportions shown in Fig. 14, which is the span

taken as the example in considering strains, etc., we have for uniform load

$$D = \frac{1.166 Wl^3}{Eh_1^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$= \frac{Wl^3}{161,500,000I}, \quad . \quad . \quad . \quad . \quad (3a)$$

and for load at end

$$D = \frac{3.377 Pl^3}{Eh_1^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$= \frac{Pl^3}{55,700,000I}. \quad . \quad . \quad . \quad . \quad . \quad (4a)$$

Where the girders have the proportions as given in Fig. 15, for uniform load

$$D = \frac{1.315 Wl^3}{Eh_1^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$= \frac{Wl^3}{132,000,000I}, \quad . \quad . \quad . \quad . \quad (5a)$$

and for a load at end

$$D = \frac{4.248 Pl^3}{Eh_1^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$= \frac{Pl^3}{40,800,000I}. \quad . \quad . \quad . \quad . \quad . \quad (6a)$$

W and I as given above. Some one of the formulæ would be applicable to any case likely to occur.

Considering first the case of uniform load: the girder we have been considering is composed of $84'' \times \frac{3}{8}''$ web, four $5'' \times 3\frac{1}{2}'' \times \frac{9}{16}''$ angles, and (neglecting the short splice-plate), two $12'' \times \frac{1}{2}''$ plates in top flange and two $12'' \times \frac{5}{8}''$ in bottom flange. To simplify the calculations, we will for the present consider all cover-plates as $\frac{1}{2}$ in.; if this is not done, we should first find the centre of gravity of the section, and then the moment of inertia about this axis. Usually the flange-plates

are the same, and we will obtain nearly correct results by so considering them. The moment of inertia of the web about its centre is equal to $\frac{1}{12}bh^3$. ($b = \frac{3}{8}$, and $h = 84$.) $\frac{1}{12}bh^3 = \frac{1}{12} \times \frac{3}{8} \times 84^3 = 18,522$. The moment of inertia of the cover-plates and angles about the centre of the web is found by multiplying the area of each by the square of the distance between its centre of gravity and the centre of the web. Thus the area of the four $\frac{1}{2}$-in. plates $= 24$ sq. in., and the square of the distance from the centre of web to their centre is $(42 + \frac{1}{2})^2$. $24 \times (42.5)^2 = 43,350$. By referring to Carnegie's Pocket-book we see that the centre of gravity of the $5 \times 3\frac{1}{2}$ angles is about 1 in. from the back of the angle, and that the area of the four angles is 17.88 sq. in. The half-depth 42 in. — 1 in. gives 41 in. as the distance from the centre of the web to the centre of gravity of the angles. $17.88 \times (41)^2 = 30,056$. To the moments of inertia thus obtained we add the moments of inertia of the cover-plates, and the angles about their own centres of gravity; for the cover-plates $\frac{1}{12}bh^3 = \frac{1}{12} \times 12 \times 1$ in. $= 1$ for each flange, and for the angles we have from the Pocket-book 4.2 for each angle (see page 103, edition of 1893). $4.2 \times 4 = 16.8 + 2 = 18.8$, amount to add for plates and angles. The total moment is then $18,522 + 43,350 + 30,056 + 18.8 = 91,946.8$. It will be noticed that the moments of inertia of the plates and angles about their own axis is very small, and might be neglected without seriously affecting the result.

Using our formula No. 3*a*, we have

$$D = \frac{Wl^3}{161,500,000I}.$$

$W = 22,300$, as previously found, $L = 42.5$ ft., and $I = 91,946.8$.

$$D = \frac{22,300 \times 132,651,000}{161,500,000 \times 91,946.8} = 0.19 \text{ inch} = \tfrac{3}{16} \text{ inch}.$$

If each arm is given a camber, this must be considered in determining the end deflection. Suppose the top chord be lengthened by adding $\frac{1}{4}$ in. at a web-splice near the centre of the arm. If the girder be 5 ft. 6 in. deep at this point, and the distance to the end be 21 ft., the end will drop $\frac{1}{4} \div 5.6 \times 21 = .94$, say $\frac{61}{64}$ in. Adding $0.19 + .94 = 1.13$ in. $= 1\frac{1}{8}$ in., end deflection.

MACHINERY.

For Turning.—The forces to be overcome in turning the draw are, first, the inertia of the span itself. That is, there is a certain mass which has to be revolved through a quarter of a circle or 90° of an arc in a certain time. Second, there is the friction on the centre pivot or rollers. Third, the friction of the trailing-wheels due to the overturning force of the wind, and the friction on the vertical surface of the pivot due to the wind-pressure. Fourth, there is the friction of the trailing-wheels due to any unbalanced load there may be. Fifth, the friction of the shaft-bearings, etc. Item four might be considerably increased by the rails on which the wheels bear being out of level, rough, and with wide openings at the joints. It is sometimes assumed that the draw shall turn against a wind-force acting on one arm only of the span. While this might possibly happen in the case of a long span, it could hardly occur in the short 85-foot span we are considering, and this condition will not therefore be treated at present.

Force required to Overcome Inertia.—For convenience we replace the mass of the bridge by an equivalent mass acting at the rack-circle. This mass is found as follows: Multiply the weight of the span by the square of half the length plus the square of half the width, and divide by 96.6 times the square of the radius of the rack-circle. Putting this in the form of an equation,

$$M = \frac{W(a^2 + b^2)}{96.6R^2},$$

where W = weight of span;

a = half-length of span;

b = half-width of span;

R = radius of rack-circle;

M = equivalent mass at rack-circle.

The weight of our span is 89,200 lbs. = W. a, the half-length, = 42.5 ft.; b, the half-width, = 3.5 ft.; and R, the radius of the rack, = 7.85 ft. We have therefore

$$M = \frac{89,200 \times (42.5^2 + 3.5^2)}{96.6 \times 7.85^2} = 27,224.$$

If we assume that the draw shall open in two minutes, the average velocity will be one fourth the circumference of rack divided by 120 sec. $= \frac{49.32}{4 \times 120} = 0.103$ ft. per second. But the velocity is not uniform; it increases during the first half of the turning, and then reduces to 0 again at the end. The maximum velocity at the end of 60 seconds is then twice the average, or 0.206 ft. per second. The rate of increase is $0.206 \div 60 = .0034$.

The force necessary to give a mass of 27,224 lbs. a constantly increasing velocity of .0034 ft. per second = 27,224 $\times$ 0.0034 = 92.5 lbs. We will call this Fm.

Force to Overcome Friction on Centre Bearing.—A Sellers centre is used so the friction from load will be rolling friction; a coefficient of .003 may be used, and this multiplied by the load gives 89,200 $\times$ .003 = 267.6 lbs. This acts at the centre of the length of the roller, or with a leverage of 8 in. or .62 ft. $267.6 \times .62 \div 7.85 = 21.1$ lbs. the force required at rack to overcome it. This force we designate F_p.

Friction on Side of Pivot or End of Rollers for Wind-pressure.—Assuming a wind-load of 300 lbs. per lineal foot,

there results a total horizontal force of $300 \times 85 = 25,500$ lbs. This, whether acting against the ends of the rollers or on the side of a pivot, will produce sliding friction. Using a coefficient of 0.1, this gives $25,500 \times 0.1 = 2550$ lbs. acting at the end of roller or at circumference of pivot (acting on vertical surface). Let the radius of end of roller be $9\frac{1}{2}$ in. or .8 ft., then $2550 \times .8 \div 7.85 = 259.8$ lbs. at rack. We will denote this by Fw.

Force required to Overcome an Unbalanced Condition of the Draw.—Suppose that from snow or some other cause there is an unbalanced load on one arm, acting at a point 15 ft. from the centre pivot. The force at the wheel-circle required to balance this is $15 \div 7$ (the radius of the wheel-circle) $= 2.143$ times the load. Assume the load to be 2000 lbs.; this multiplied by 2.143 gives 4286 as the pressure on the balance-wheel. The friction caused by this pressure will be rolling friction and equal to $4286 \times .003 = 13$ lbs. Thirteen pounds at the wheel-circle will require $13 \times 7 \div 7.85 = 11.6$ at the rack to overcome it (7 and 7.85 being the radii of the two circles). This force we will call Fu.

The centre of the surface exposed to the wind, including ties and guard-rails, is almost exactly in line with the bottom of the cross-girders, so that the moment of the wind-force tending to revolve the girders about the centre casting as a fulcrum is in this case slight and may be neglected. Suppose the centre of wind-pressure had been one foot above the point of support for cross-girders, the overturning moment would then have been $25,500 \times 1 = 25,500$ lbs.; this divided by the horizontal distance from the centre support to the centre of the trailing-wheel, 7 ft., gives the vertical force acting at wheel to resist overturning. $25,500 \div 7 = 3643$ lbs. Using coefficient of friction .003 gives 10.9 lbs. $10.9 \times 7 \div 7.85 =$, say, 9.7, force at rack necessary to overcome it. This will show how to proceed in cases where this overturning force of the wind is too great to be neglected.

Force required to Overcome the Friction of the Shaft.—There will be only one shaft required in the turning arrangement.

Assuming one man is able to turn the draw, and that he exerts a pressure of 75 lbs. horizontally against the top of the shaft; assuming for the present also that he works at the end of a five-foot lever, and that a pinion 8 in. in diameter can be used in rack, we have a horizontal pressure at foot of shaft of $75 \times 60 \div 4 = 1125$ lbs. $1125 + 75 = 1200$ lbs., total pressure on shaft-bearings. The friction caused by this will be sliding friction, for which the coefficient is 0.05 to 0.1. Multiplying $1200 \times 0.1 = 120$ lbs. as the frictional force acting at the circumference of the shaft. This we will call Fs.

We have then forces to be overcome as follows: $Fm = 92.5$, $Fp = 21.1$, $Fw = 259.8$, $Fu = 11.6$, and $Fs = 120$ lbs.

First we will see how much power is consumed in overcoming Fs. The radius of the shaft will be assumed as $1\frac{1}{4}$ in. for the present, then $120 \times 1\frac{1}{4} \div 60 = 2.5$, the power required at end of turning-lever to balance it. This leaves us $75 - 2.5 = 72.5$ lbs. as available against the other forces which all act at rack-circle. These equal $92.5 + 21.1 + 259.8 + 11.6 = 384.9$ lbs. Dividing 384.9 by 72.5 gives 5.3, which is the number of times the power must be multiplied between the turning-lever and the pinion, or by the two. We see at once that our power will be greatly in excess of the amount required. It will multiply as many times as the radius of the pinion is contained in the length of the turning-lever, $60 \div 4 = 15$ (using an 8-in. pinion). We might use a six-inch pinion and four-foot turning-lever. It is well, however, to have a good excess of power, as machinery may get out of adjustment, the track become rough, and with gaps at the joints, the span may become badly unbalanced, etc.

Time for Turning.—The man turning the draw will walk at an average velocity of, say, 3 ft. per second. If he be moving at the end of a five-foot lever, he will move in a

circle of 31.6 ft. circumference. It will require $31.6 \div 3 = 10.5$ seconds for him to make one complete revolution. The pinion of course makes one revolution in the same time. Using a pinion of 25 in. circumference on the pitch-line, and a rack of 49.3 ft. circumference, the pinion must make $\dfrac{591.6 \text{ in.}}{4 \times 25} = 5.9$ revolutions in moving over one fourth of the circumference of the rack, which would be necessary to open the draw. If one revolution is made in 10.5 seconds, $10.5 \times 5.9 = 62$ seconds as the time required to open or close the draw.

Size of Turning-shaft.—The man moving at the end of the turning-lever produces a twisting moment on the shaft of $75 \times 60 = 4500$ in.-lbs. In addition to this twisting there is the bending produced by the force acting on the pinion.

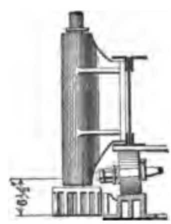


Fig. 16

Assuming an 8-in. pinion, this force equals 1125 lbs.; and assuming that the lower corner of the tooth is acting, and that the distance from this corner up to, say, $1\frac{1}{2}$ in. inside the journal-bearing equals $6\frac{1}{2}$ in., then the bending moment will be $1125 \times 6\frac{1}{2} = 7312$ in.-lbs. By referring to the notes on shafting we find that the strength of a shaft to resist both bending and twisting is given by the formula

$$T' = M + \sqrt{M^2 + T^2}.$$

$M =$ bending moment, and $T =$ twisting moment.

$$T' = 7312 + \sqrt{53465344 + 20250000} = 7312 + 8585 = 15897.$$

Adding 50 per cent to this to allow for contingencies, we have 23,846 in.-lbs., requiring a 2⅜-in. diam. shaft. Note that the shaft is weakened by the keyways and the shoulders for turning-lever.

Proportions of Trailing-wheels.—The face of the wheel should be about 4 in., to make sure it always has bearing on the rail and to keep the bearing back from the edge. Letting w = width of face, the other proportions would be about as follows: Thickness of rim = $.4W$; thickness of solid web = $.25w$; stiffening-ribs, six in number, thickness = $.2w$; length of hub, not less than $1.5w$; diameter of hub, 1.85 times the size of axle required.

The side bearings should not be less than the diameter of the axle, giving total bearing of $2D$ or more.

In figuring the size of axle required, if a length from the centre of the wheel to the centre of bearing be used, the unit stress in bending might be assumed at 30,000 lbs. per square inch. The reason for this is that the bearings and hub practically fix the axle so that it cannot bend until it leaves the hub or the bearings.

Strength of Teeth in Rack and Pinion.—Referring to the tables and notes on the strength of teeth, we find the formula for the safe load on cast-iron teeth $P = 375t^2$. This formula is for the strength of tooth considering the load as applied at one corner. We found the pressure on the tooth to be 1125 lbs.; then $P = 1125 = 375t^2$. $t^2 = 3$, and $t = 1.73$. (See table of cast-iron teeth.) We find also from the table that the width of face must be 2½ in. to give the same strength, assuming the load as uniformly spread over the length of face. As the speed is slow, we use the value of P_1 for 100 ft. per minute or under. It is common practice to make the breadth of the tooth not less than two to three times the pitch.

Steel Rollers in Centre Bearing.—Making the rollers hard steel on hard-steel bearing-plates, we can allow a pres-

sure per lineal inch of roller of $1750 \sqrt{d}$; d being the average diameter of roller. Calling this average diameter 2.5″, we have 2765 lbs. allowed pressure per lineal inch. The weight of the span is 89,200 lbs., and this divided by 2765 gives 32.2 lineal inches required. There are 15 rollers, 3 in. long, giving 45 in. actual.

If a centre-pin is not used, care should be taken to give the ends of the rollers an even bearing to resist the lateral pressure as explained above. The plates or rings between which the rollers move should be thick enough to distribute the pressure evenly and so that there will be no give or spring as the span revolves. For three-inch rollers the plates should not be less than $2\frac{3}{4}$ to 3 in. thick.

If a pivot with flat disks had been used (see details of this form of centre), the coefficient of friction would have been about 0.1 (see table of allowed bearing on disks of steel and bronze). The centre of pressure on pivots is at two thirds the radius from the centre.

Wedging Arrangement at Centre and Ends.—The centre roller-bearing is supposed to carry dead load only. To support the span under live load, wedges or some equivalent device are used under the girders at the centre and at the ends. The supports at the centre should be driven just hard enough to bring them to a full solid bearing, but not hard enough to take the dead load off the centre pivot or rollers. The amount the end wedges should drive is determined by the amount of deflection it is found necessary to take out of the girder so that there shall be no raising of the ends off the supports as the load passes over one arm. The gears or levers moving the wedges are easily arranged to give any desired amount of motion to either set. The amount it is necessary to raise the ends of the girder will now be considered. Placing the engine on one arm with the heavier wheels at the centre, we find the reaction at the end of unloaded arm to be 7070 lbs. (see Fig. 9.) This means that

a force of 7070 lbs. must be applied at the end of unloaded arm to prevent its raising off the support. This force may be obtained by driving the wedges under the ends of the girder, and giving it an upward deflection until it is strained sufficiently to give the reaction required.

Our formula for the deflection from an end load and girder of varying section is, from page 26, No. 4*a*,

$$D = \frac{Wl^3}{55,700,000I}.$$

We have $W = 7070$ lbs., $l = 42.5$ ft. $= 510$ in., $I = 91,946.8$.

$$D = \frac{7070 \times 132,615,100}{55,700,000 \times 91,946.8} = .185 = \tfrac{3}{16} \text{ inch.}$$

Our wedge must then have a vertical movement of something over $\frac{3}{16}$ in. If we make the slope of the wedge 1 in 5, a horizontal throw of 12 in. will give us ample clearance for turning.

The horizontal force necessary to drive the wedge will be $\frac{7070}{6}$ ($\frac{1}{6}$ being the slope of the wedge) plus the friction of the top and bottom surfaces of the wedge on their bearings. This friction we will assume as 236 lbs. Then $\frac{7070}{6} + 236 = 1414$, which is the horizontal force to be applied. The coefficient of friction might be as high as 0.10. At this value we have $\frac{7070}{6} + 707 + 707 = 2592$ lbs. as against the 1414 lbs. we are now using. It will be noticed that the friction is an important element in determining the actual power to be derived from the wedge.

The centre wedges should not be driven hard enough to lift the span off the centre support, but just to a solid bearing. We will assume, however, for the present that all six wedges are driven with a force of 1414 lbs. each. This will give us an excess of power of about 50 per cent. One man, it was assumed, could exert a force of 75 lbs. The power must then be multiplied between the man and the wedges. $1414 \times 6 = 8484 \div 75 = 113.2$ times. Using a 60-inch lever and

the worm-screw arrangement as shown in Fig. 46, in one revolution of the shaft the man moves $120 \times 3.14 = 376.8$ ft. The pitch of the screw is, say, $2\frac{1}{2}$ in., or there is a vertical motion of $2\frac{1}{2}$ in. Dividing 376.8 by $2\frac{1}{2}$ gives 150.7 as the multiplication of power, against 113.2 required. We do not then need to increase the power further, and all arms on the shafts may be of the same length. If the rods connecting centre and end shafts are on one side of the bridge only, that is, if one set only are used (sometimes one and sometimes two are employed; if the bridge is wide, there should be a set on each side), these rods will carry a strain of $1414 \times 2 = 2828$ lbs. each. Rods $\frac{5}{8}$ or $\frac{3}{4}$ in. round will be ample. The worm-shaft has a twisting moment of $75 \times 60 = 4500$ in.-lbs.; by the table on shafting we see that this requires a shaft of, say, $1\frac{7}{16}$ in. diameter. In order to make a suitable thread for the worm, the shaft ought not to be less than $5\frac{1}{2}$ or $5\frac{3}{4}$ in. diameter. So in this case the worm would determine the size of shaft to use.

The angle of repose for steel on cast iron is, say, 11°. The thread of the worm should then have a slope not exceeding 10° or 12°. If the pitch is $2\frac{1}{2}$ in., the thread rises $1\frac{1}{4}$ in. in one half-revolution, and the angle is found by dividing this rise ($1\frac{1}{4}$ in.) by the diameter of screw on the pitch-line. Assuming this to be 5.8, we have $1.25 \div 5.8 = .21 =$ tangent of 12°. Rather than use a shaft of this diameter, it would be better to make the worm in the form of a sleeve, and key it to a $2\frac{1}{4}$ or $2\frac{1}{2}$ in. shaft. Or a shaft $3\frac{1}{2}$ or $3\frac{3}{4}$ in. in diameter might be used with a worm of $1\frac{1}{4}$ or $1\frac{1}{2}$ in. pitch. The objection to this arrangement for such a light span is that the time required to operate the machinery is made unnecessarily great. We will assume that the worm is made in the form of a sleeve and has a diameter at the centre of the thread (or pitch-line) of 5.8 in.

Horizontal Shafts.—We found that we multiplied our power between the end of the turning-lever and the sliding-

or worm-nut on the vertical shaft 150 times. The force exerted by one man at the end of the turning-lever was assumed as 75 lbs. Then $75 \times 150 = 11,250$ would be the force exerted upon the sliding-nut, were not a portion of this used in overcoming the friction of the various parts. We will first determine what these frictional forces are, up to the point where the nut-lever acts on the horizontal shaft. These forces being found and subtracted from 11,250 will give us the force that the horizontal shaft must carry on to the wedges.

We have, first, the friction of the bearings of the vertical shaft; second, the friction on the collars from the thrust of the vertical shaft; third, the friction of the sliding-nut in its guides; and fourth, the friction of the sliding-nut on the thread of the worm-shaft.

These are all sliding frictions for which the coefficient would be between 0.05 and 0.1, depending upon the smoothness of the surfaces and the amount and character of the lubrication. We will use 0.06.

The horizontal pressure on the journals is the 75 lbs. exerted by the man at the lever increased by the leverage due

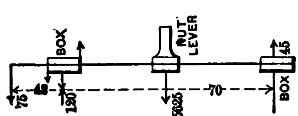


Fig. 17

to the bearing being some distance below the lever. Suppose the lever to be 42 in. above the box, and that the play in the box is sufficient so that the lower box might be assumed as resisting this bending; then we have $75 \times 42 \div 70 = 45$, as bearing on lower box. There is also the horizontal pressure from the worm-nut in its guides. This is equal to $75 \times 60 \div 8 = 562.5$ lbs. (Eight inches being the distance from the centre of the shaft to the centre of bearing of the nut on

its guides.) Forces causing friction on the bearings are then $120 + 45 + 562.5 = 727.5$ lbs. If we use a $2\frac{1}{2}$-in. shaft,

$$727.5 \times .06 \times 1.25 \div 60 = .91, \quad . \quad . \quad . \quad (1)$$

the force at end of lever to overcome this friction (1.25 being the radius of the shaft, and 60 the length of the hand-lever).

Friction of the Collars.—The vertical thrust on the shaft we found to be 11,250 lbs. This acts on the collars with a leverage (the distance to the centre of gravity of the ring) of, say, $1\frac{3}{4}$ in.; then $11,250 \times .06 \times 1\frac{3}{4} \div 60 = 19.7$, the force at end of hand-lever. This is excessive, and the friction should be reduced by using a ball bearing in the collars (see detail of this arrangement in cuts). This reduces the friction to rolling instead of sliding friction, and the coefficient to .003; we have then

$$11,250 \times .003 \times 2 \div 60 = 1.12. \quad . \quad . \quad . \quad (2)$$

Friction of Worm-nut Sliding in its Guides.—The horizontal pressure of the nut we found to be $75 \times 60 \div 8 = 562.5$. Then

$$562.5 \times .06 \div 150 = 0.22. \quad . \quad . \quad . \quad . \quad (3)$$

(The number 150 is the number of times the power is multiplied between the hand-lever and the nut.)

Friction on the Worm-thread.—The vertical pressure is 11,250; and if the slope of the thread is 12°, this gives a force in the direction perpendicular to the screw-thread of $11,250 \div 1.022 = 11,008$. $11,008 \times .06 = 660.48$.

We will assume that the force at end of hand-lever necessary to overcome this friction is 4.4 lbs. This force is equal to the friction multiplied by the radius of the worm-thread, divided by the length of the hand-lever. In some cases the friction may reduce the efficiency of the worm 40 to 50 per cent. (See page 86.)

The sum of these frictions is $.91 + 1.12 + 0.22 + 4.4 = 6.65$ lbs. Subtracting this from 75 gives $75 - 6.65 = 68.35$, the available power at hand-lever. 68.35×150 (the number of times power multiplies) $= 10,253$, the power transferred by worm-nut to the arms on the horizontal shaft.

The horizontal shafts have, in addition to the twisting moment, the bending due to the distances between the bearings and the various levers which are keyed to the shafts. On the centre shaft we have the levers or arms working the struts which draw the centre wedges, the arms driving the rods to the end wedges, and the arms working into the worm-nut.

On the end shaft we have the arms working the end wedges, arms worked by long rods running to centre, and the cranks which work the rail-lifts. The twisting moment extends nearly uniformly through the centre shaft if the centre wedges are only driven to a bearing, and there are rods running to the end shafts on each side of the bridge. If the rods are on one side only, the moments of the twisting force will be greatest between the worm-nut lever and the end of shaft carrying the rod-arms.

In the end shaft, with one set of driving-rods, the moment is greatest between the arms driven by the long rods and the strut driving the end wedge. Then it is reduced by the amount of the moment on the wedge strut-arm. It is again reduced by the amount of rail-lift moment when this point has been passed, and so on to the other end.

With two sets of the driving-rods the moment at the centre would be 0, and increase each way to the ends. For the bending moments the portion of shaft between bearings will be considered as a single span, and the bending moments in each portion combined with the twisting moment (see table and formulæ for shafts).

The distance from one arm or prong of the lever working in the worm-nut to the nearest bearing is, say, 8 in., and as

each prong carries half the load, the bending moment will be $10,253 \div 2 \times 8$ in. $= 41,012$ in.-lbs. The twisting moment is, if there are driving-rods on each side, $5126.5 \times 11 = 56,391$ in.-lbs. (11 being the length of the arm or prong from the centre of the shaft). If the driving-rods are on one side only of the bridge and run from the centre to the end on opposite sides, for opposite ends as in Fig. 19, the moments

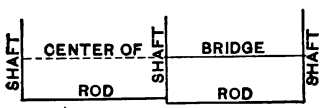


Fig. 18

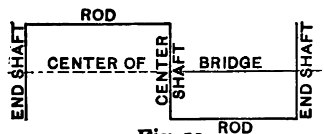


Fig. 19

are the same; but if the rods are on the same side, as in Fig. 18, the moment will be $10,253 \times 11 = 112,783$.

The first arrangement should of course be used, and we have bending moment $= 41,012$ and twisting moment $56,391$. Our formula (see notes on shafting) is

$$T' = M + \sqrt{M^2 + T^2}.$$

$$T' = 41,012 + \sqrt{4,841,928,825} = 41,012 + 69,584 = 110,596.$$

By the table a shaft of 4 in. diameter is required for this moment.

The bearings should be placed as near the points of loading as possible.

The bearings of the horizontal shaft at the centre of the bridge carry a pressure of twice the vertical force at nut-lever, or 20,506 lbs. Using a coefficient of 0.06, and remembering that the lengths of all levers on this shaft are 11 in., we have power lost in friction on this shaft $20,506 \times .06 \times 2 \div 11 = 223.7$ lbs., and the shafts at the ends of bridge, including rail-lifts, have about the same amount (it would be figured in precisely the same manner). $10,253 - (224 + 224) = 9805$

lbs., available for driving wedges, or 1634 lbs. to a wedge against 1414 required. As the machinery is liable to get out of adjustment and the bearings to become dry, there should be at least 100 per cent excess of power. The wedges will

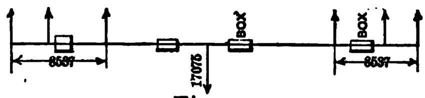


Fig. 20

stick and more power will be required to start them than will be necessary to move them when once started.

Special care should be taken to provide ample means for lubricating the wedges. The surest method is perhaps to make several deep grooves diagonally across the bearing-surfaces. These grooves will retain a large amount of oil and, as the wedges move, spread it over the surfaces. All oil-holes should be easy of access and provided with means for excluding dirt.

The Levers.—The lever-arms and the wings on the sliding-nut should be figured as beams fixed at one end and

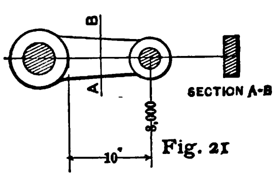


Fig. 21

loaded at the other. For cast iron the fibre-stress should be about 4000 lbs., and for cast steel 15,000 lbs. The bending moment divided by the fibre-stress gives the momennt of resistance required; thus $R = \frac{M}{f}$. M = bending moment, f = fibre-stress, and R = moment of resistance. Say we have a pull of 8000 lbs. at the end of an arm, and the distance to the points where the arm joins the hub is 10 in.; the

moment (M) is then 80,000 in.-lbs. if the arm is cast iron. $M \div f = 80,000 \div 4000 = 20$. Using a rectangular section, $R = \frac{1}{6}bh^2$ (see any table on moments of resistance and inertia). Assuming $b = 1.25$, then $20 = \frac{1}{6} \times 1.25 \times h^2$. $h = 2.1$. If a rectangular section is used, it should be stiffened by ribs on the sides if the length exceeds six or eight inches. It must be remembered that these levers are subject to sudden jars, and should be made amply strong. The hubs are weakened by the keyways, and should not be less than $1\frac{1}{4}$ to $1\frac{1}{2}$ in. thick. The keyways should be cut in the side of hub next the arm where there is the greatest excess of metal. A table giving the common sizes of keys used in shafts of different diameters is given below.

Elbow-joint.—We have found that our power has been amply multiplied by the turning-lever and the worm. It may be, and in fact the arrangement of crank on horizontal shaft and the strut driving-wedge should be, such that they increase the power two or more times. When the wedges are driven the crank and strut should be in the same straight line, or nearly so. As the force on the crank acts tangentially to the circle described by its end pin, when the crank and strut are nearly in line this tangential force is capable of exerting an enormous pressure in the direction of the strut. As the angle between the strut and crank increases this force decreases. It will be noticed that when the crank and strut are nearly in line there may be a movement of the crank through a considerable arc and very little movement in the direction of the strut, so that to get the necessary amount of action in the wedge the crank must move to a position where it is not acting to the best advantage. Assuming that the power necessary to drive the wedge increases regularly from 0 at the point where the wedge just takes a bearing to a maximum when the wedge is fully driven, then Figs. 11 to 14 and the explanation below them show how a few trials with the wedge driven to different positions will determine in which one of

them the greatest tangential force is required. And this greatest force is the one to be used in determining the moments on the shaft and the power required to turn.

If, when the wedges are driven, the crank and strut stand at a considerable angle, there may be danger of the wedge working loose under the action of live load, especially if the angle of the wedge is steep.

Example of Elbow- or Toggle-joint.—Assume that the horizontal force necessary to raise the end of the girder the required amount be 2000 lbs. moved through a distance *AB*. The horizontal force is zero when the wedge is drawn out, so that the point *A* coincides with the point *B* and increases as the distance between *A* and *B* increases. Assume that when

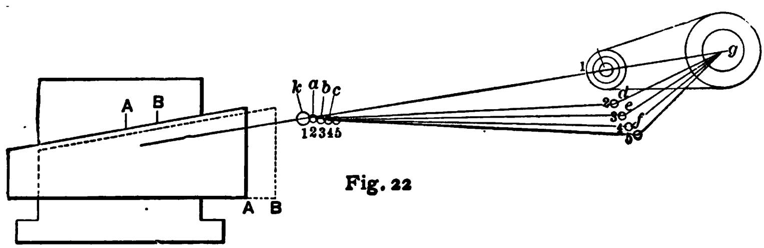


Fig. 22

the wedge is driven the line of the crank and strut will be *ghk*. With wedge moved $\frac{1}{4}$ of *AB* line of lever and strut, assume line *adg*. Moved $\frac{1}{2}$ of *AB*, the line becomes *beg*.

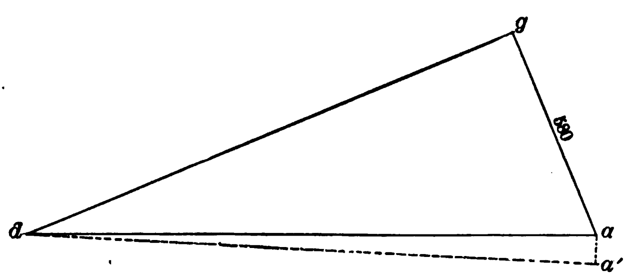


Fig. 23.

Moved $\frac{3}{4}$ of *AB*, the line becomes *cfg*. To find tangential force at end of crank, with end of crank at *d*, lay off from *d*

(Fig. 23) a line parallel with *ad*, also a horizontal line on which lay off the force on wedge at this point = ¾ of 2000 = 1500. From *a′* draw a perpendicular to *da′*, intersecting *da* at *a*; through *d* draw *dg*, parallel with *dg* in Fig. 22. Through

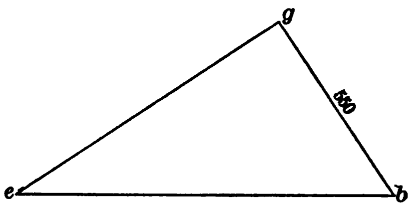


Fig. 24.

a draw *ag* at right angles to *dg*. Line *ag* equals the tangential force at *d*. Figs. 24 and 25 are drawn in the same way and give the tangential force at *e* and *f*.

The Latch.—Several styles of latch are given in the cuts. The object of the latch being to hold the bridge in exact line, it should fit close when driven to place, and it must be strong

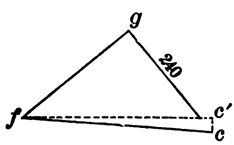


Fig. 25.

enough to hold the bridge against the wind; and if it acts automatically, it must resist the shock of stopping the bridge suddenly as it swings in position. Sometimes the latch drives at the same time as the wedges are driven, but a latch working independently is more satisfactory.

Rail-splices. — The latch should not be relied upon entirely to keep the rails in line, but a sleeve of some sort, slipping over the ends of the rails both on the draw and the abutment, should be used.

Signals.—The levers working the latch or the wedges may also throw danger-signals placed on the abutments, or the

span as it revolves may be made to throw them. If there are many attachments to the same set of machinery, some of them are pretty sure to be out of adjustment most of the time. And in general the simpler the machinery of a draw-span is the better. A few heavy, amply strong parts are infinitely better than a great mass of light, complicated pieces; the one will be satisfactory in service, the other never will be.

Set-screws.—While set-screws may be used in places where there is little stress, they are not satisfactory in most places on draw-span machinery. When most needed they can only be relied upon *to fail.* Where used they should be not less than $\frac{3}{4}$ or $\frac{7}{8}$ in. diameter. If two are used at one connection, they should not be placed opposite to each other, but at right angles.

Care of Draw-spans.—To give satisfaction, the best designed draw must have constant care and attention. Many complaints of spans not working satisfactorily are due to gross neglect in their treatment. The writer once went to a draw that was giving trouble, and found that a coil of old rope left on the pier some months previously by bridge-carpenters had become wedged in between the rack and the pinion and wrapped around the shaft, rendering it almost impossible to turn the draw. How often had this part of the machinery been examined in that time? Not once. In fact, some parts out of sight and not easy of access had not been oiled in a year or more. The surest way to insure care in this respect is to have as few parts as possible, and these easy to be seen and reached. Other things being equal, the best design is the one with the fewest parts to keep in repair.

TABLES AND GENERAL DATA.

Notes on Spur- and Bevel-gears.

PROPORTIONS OF TEETH.

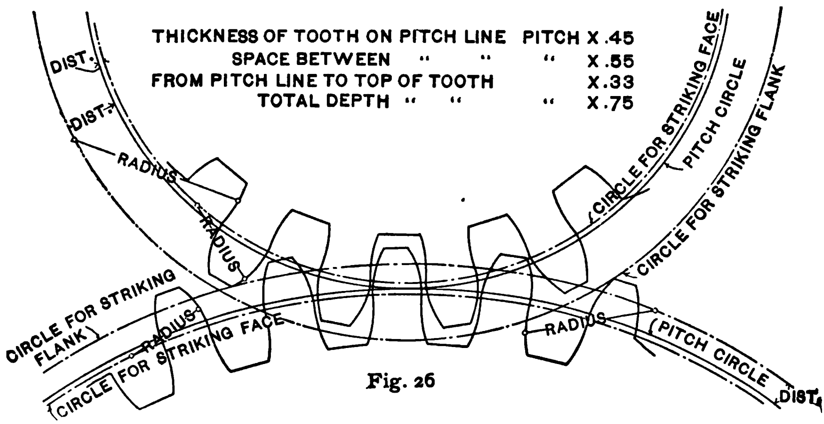


Fig. 26

GRANT'S ODONTOGRAPH TABLE FOR EPICYCLOIDAL TEETH.

Number of Teeth.		For One Diametric Pitch. For any other pitch diameter divide by that pitch.				For 1″ Circular Pitch. For any other pitch multiply by that pitch.			
		Face.		Flank.		Face.		Flank.	
Exact.	Interval.	Rads.	Dist.	Rads.	Dist.	Rads.	Dist.	Rads.	Dist.
12	12	2.01	.06	∞	∞	.64	.02	∞	∞
13½	13 to 14	2.04	.07	15.10	9.43	.65	.02	4.80	3.00
15½	15 " 16	2.10	.09	7.86	3.46	.67	.03	2.50	1.10
17½	17 " 18	2.14	.11	6.13	2.20	.68	.04	1.95	.70
20	19 " 21	2.20	.13	5.12	1.57	.70	.04	1.63	.50
23	22 " 24	2.26	.15	4.50	1.13	.72	.05	1.43	.36
27	25 " 29	2.33	.16	4.10	.96	.74	.05	1.30	.29
33	30 " 36	2.40	.19	3.80	.72	.76	.06	1.20	.23
42	37 " 48	2.48	.22	3.52	.63	.79	.07	1.12	.20
58	49 " 72	2.60	.25	3.33	.54	.83	.08	1.06	.17
97	73 " 144	2.83	.28	3.14	.44	.90	.09	1.00	.14
290	145 " rack	2.92	.31	3.00	.38	.93	.10	.95	.12

SPUR GEAR.

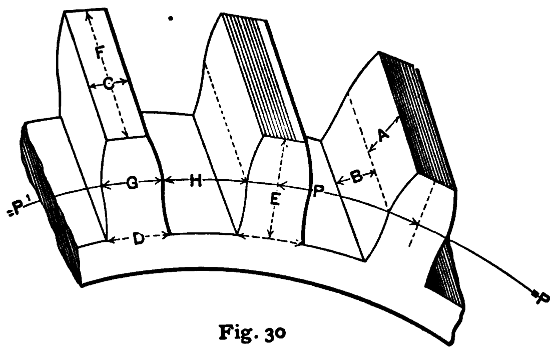


Fig. 30

$P'P'$ = pitch-circle;
A = face;
B = flank;
C = point;
D = root;
E = height;
F = breadth;
G = thickness;
H = space;
P = circular pitch.

RACK.

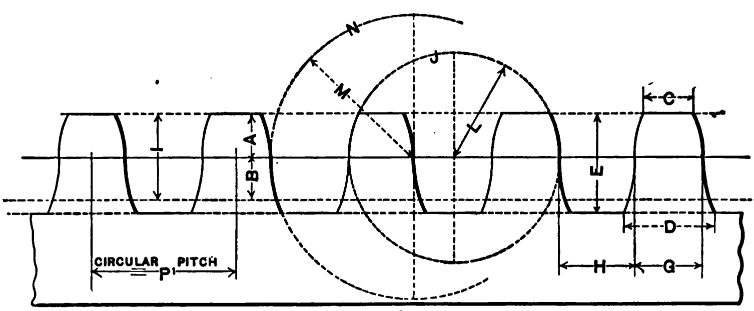


Fig. 31

Double-curve Teeth for Racks and Wheels.

Circle J for face-radius $L = P' - \frac{1}{2}$ of G. Circle N for flank-radius $M = P'$.

BEVEL AND MITER GEARS.

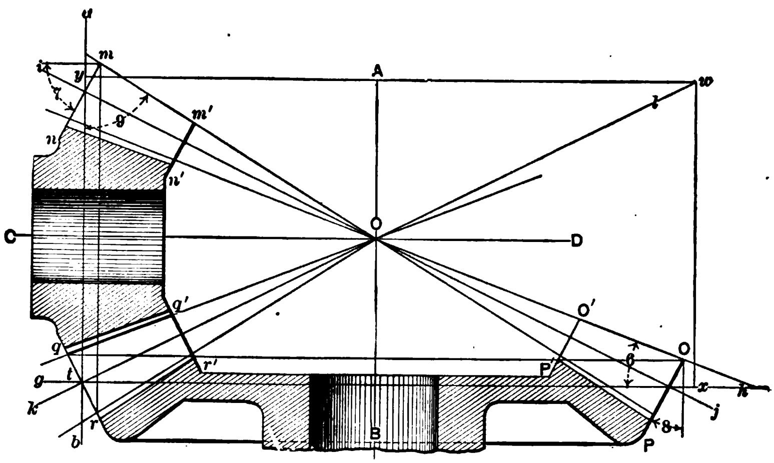


Fig. 32.

AOB = centre of wheel;
COD = " " pinion;
ab = largest pitch diameter of pinion;
gh = " " " " wheel;.
OiC and OkC = angles of cone pitch-line of pinion;
OjB and OkB = " " " " " " wheel;
mr = whole diameter of pinion;
qO = " " " wheel;
wt and iO = working depths of tooth.
$\frac{1}{10}$ of $wy + ab = mr$;
$\frac{1}{10}$ of $wx + gh = qO$.
angle 9 = angle of face of pinion;
angle 6 = angle of face of wheel.

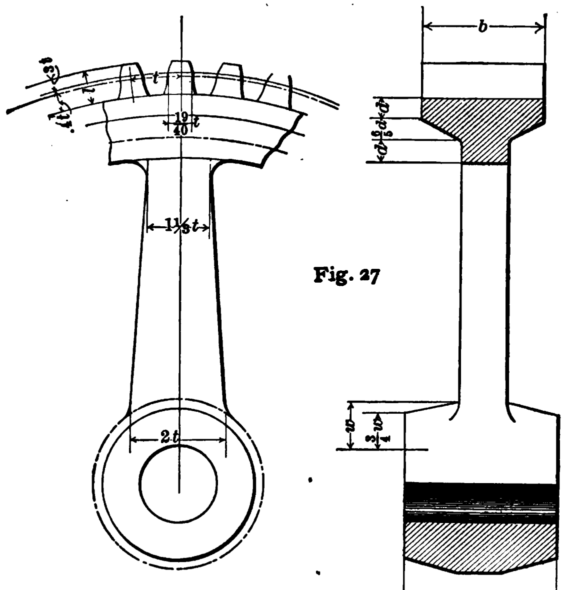


Fig. 27

$d = 0.4t + 0.125''$
$w = 0.4h + 0.4''$

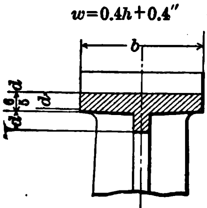


Fig. 27ᵃ

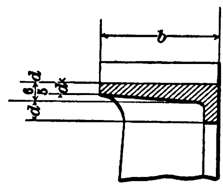


Fig. 27ᵇ

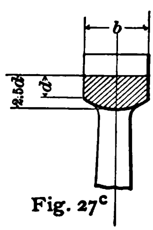


Fig. 27ᶜ

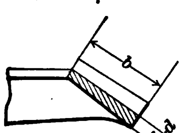


Fig. 27ᵈ

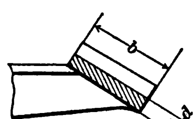


Fig. 27ᵉ

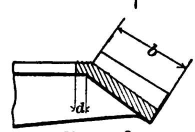


Fig. 27ᶠ

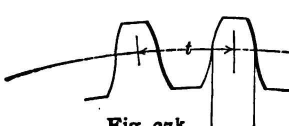


Fig. 27ᵏ

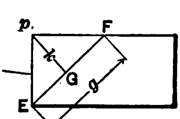


Fig. 28

Gear Teeth.—Cast teeth should be made sufficiently strong to resist the whole force transmitted by a pair of wheels acting on corner of one tooth, and pitch is determined as below (see Fig. 28):

Let $e =$ thickness of tooth $= \frac{19}{40}t$; $EF = g = .99t$; $PG = k = .495t$; $P =$ force at point p; moment of flexure $= Pk$; and greatest stress produced by moment of flexure on section EGF is

$$S = \frac{\text{moment of flexure}}{\text{moment of resistance}} = \frac{6Pk}{ge^2},$$

which is a maximum when angle $PEF = 45°$ and $g = 2k$. Having then the value $S = \frac{3P}{e^2}$, consequently the proper thickness for tooth is given by the equation

$$e = \sqrt{\frac{3P}{S}},$$

in which S may be taken at the values given in the table. e may be assumed to be thickness on pitch-line $= \frac{19}{40}t$; then

$$t = \frac{40}{19}\sqrt{3\frac{P}{S}}, \quad \text{when} \quad h = \frac{19}{40}t.$$

The above method of figuring the tooth is independent of the face of the tooth, and should generally be used when there is a liability of inaccuracies in the teeth.

If the face of the tooth is to be considered, as in machine-cut teeth, the pitch can be assumed and the face (b) obtained from the following.

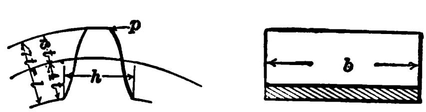


Fig. 29.

STRENGTH OF GEAR-TEETH.

Values of P_1 and P_s = the safe load assumed as applied at the corner of tooth:

$S_1 = 4500$ lbs.
$S_s = 15000$ lbs.
$P_1 = 375t^2$.
$P_s = 1250t^2$.
t = pitch.

Values of P_1 and P_s = the safe load on teeth per lineal inch of face, assuming the load as uniformly distributed over the face at outer edge, for iron and steel at varying velocities as per the table below. V = velocity in feet per minute; S_1 = unit stress for iron; S_s = unit stress for steel.

V	100 Feet or under.	200 Feet.	400 Feet.	600 Feet.	800 Feet.	1000 Feet.	1500 Feet.	2000 Feet.	2500 Feet.
S_1	4500	4310	4020	3730	3440	3150	2860	2470	2180
S_s	15000	14350	13385	12420	11450	10490	9524	8224	7260

VELOCITY IN FEET PER MINUTE.

Pitch.	Safe Load at Corner of Tooth.		100 Feet or under.		200 Feet.		400 Feet.		600 Feet.		800 Feet.		1000 Feet.		1500 Feet.		2000 Feet.		2500 Feet.	
t	P_1	P_s	P_1	P_s	P_1	P_s	P_1	P_s	P_1	P_s	P_1	s	P_1	P_s	P_1	P_s	P_1	P_s	P_1	P_s
1	375	1250	269	896	258	858	240	800	223	743	205	685	188	627	171	569	147	492	129	433
1¼	585	1948	337	1122	323	1075	301	1002	279	930	257	858	236	785	214	713	185	616	163	542
1½	844	2811	404	1345	387	1288	361	1201	335	1115	309	1028	283	942	257	855	222	738	195	650
1¾	1148	3823	470	1565	450	1499	420	1398	390	1297	359	1196	329	1095	299	994	258	859	227	756
2	1500	4995	538	1791	515	1715	480	1600	446	1485	411	1369	376	1254	342	1138	295	983	260	866
2¼	1898	6320	605	2015	579	1930	540	1800	501	1670	462	1540	423	1410	384	1280	332	1105	292	974
2½	2350	7825	673	2241	644	2146	601	2002	558	1857	514	1713	471	1568	428	1424	369	1230	325	1084
2¾	2836	9440	740	2464	709	2360	661	2201	613	2043	565	1883	518	1725	470	1566	406	1353	358	1191
3	3375	11238	807	2687	773	2573	721	2401	669	2228	617	2054	565	1881	513	1708	443	1475	390	1299
3¼	3960	13185	875	2913	838	2790	781	2602	725	2415	669	2227	612	2039	556	1851	480	1599	423	1408
3½	4594	15300	941	3133	901	3000	840	2799	780	2597	719	2395	658	2193	598	1991	516	1720	455	1515
3¾	5271	17555	1008	3358	966	3216	900	3000	836	2783	771	2567	705	2350	641	2134	553	1843	487	1623
4	6000	19980	1076	3583	1030	3432	961	3201	892	2970	822	2739	753	2508	634	2277	591	1967	520	1732

For a pitch t, face b, length of teeth l, and base thickness of tooth h, we have for a tooth-pressure p and fibre-stress S the general formula

$$bt = 6\frac{P}{S}\left(\frac{l}{t}\right)\left(\frac{t}{h}\right)^2$$

and for proportions of teeth given, h being assumed at $\frac{1}{2}t$,

$$bt = 16.8\frac{P}{S}, \quad P = \frac{btS}{16.8}.$$ (See Table, page 51

In any case the breadth of face should not be made less than $1\frac{1}{2}t$, and is generally made from $2t$ to $3t$.

It is found that the breadth of face of the tooth should increase with the increase of p. As the wear on the tooth depends on the breadth, the tooth should be proportioned so that $\frac{pn}{b}$ should not exceed a given amount. For iron $\frac{pn}{b} =$ not more than 28,000. $n =$ number of revolutions per minute.

For small forces this constant may be made as low as 12000 or 6000 without obtaining inconvenient dimensions.

For Hoisting Gears, linear velocity at pitch-circle not exceeding 100 ft. per minute, S may be taken at 42,000.

For Transmission Gears, velocity exceeding 100 ft. per minute, take S from table on page 51, in which $S = \frac{9600000}{v + 2164}$ for cast iron. For steel S may be taken $3\frac{1}{3}S$ for cast iron. $v =$ lineal velocity in feet per minute.

Arms of Gears.—A good proportion for the arms is obtained when their number A is made as follows: *

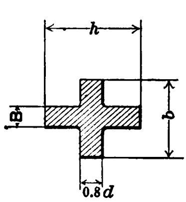


Fig. 27g

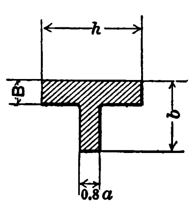


Fig. 27h

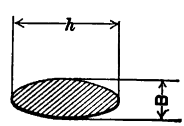


Fig. 27i

* From Releaux.

$$A = 0.53\sqrt{Z}\sqrt[4]{t}; \qquad Z = \text{number of teeth};$$

$$A = 0.73\sqrt{Z}\sqrt[4]{\frac{t}{\pi}}. \qquad t = \text{pitch}.$$

$A =$	3	4	5	6	7	8	10	12
$Z\sqrt{T} =$	30	53	83	119	162	211	330	475
$Z\sqrt{\frac{t}{\pi}} =$	11	23	36	52	71	93	146	209

Width of arm $h = 2$ to $2.5t$.

For thickness $\frac{B}{b} = 0.07\frac{Z}{A}\left(\frac{t}{h}\right)^2$.

TABLE OF GEAR-WHEEL ARMS.

$\frac{h}{t}$	Value of $\frac{B}{b}$ when								
	$\frac{Z}{A} = 7$	9	12	16	20	25	30	35	40
1.50	0.20	0.28	0.37	0.50	0.62	0.78	0.93	1.08	1.24
1.75	0.16	0.21	0.27	0.37	0.46	0.57	0.69	0.80	0.91
2.00	0.12	0.16	0.21	0.28	0.35	0.44	0.53	0.61	0.70
2.25	0.10	0.12	0.17	0.22	0.28	0.35	0.41	0.48	0.55
2.50	0.08	0.10	0.13	0.18	0.22	0.28	0.34	0.39	0.45
2.75	0.06	0.08	0.11	0.15	0.18	0.23	0.28	0.32	0.37
3.00	0.05	0.07	0.09	0.12	0.16	0.19	0.23	0.27	0.31

WEIGHT OF GEARS.

The approximate weight G of gear-wheels proportioned according to the preceding rules may be obtained from the following:

$$G = 0.0357bt^2(6.25Z + 0.04Z^2).$$

The following table will facilitate the application of the formula, as it gives the value of $\frac{G}{bt^2}$ for the number of teeth which may be given, and the weight can at once be found by multiplying the value in the table by bt^2.

	Z	0	2	4	6	8
Number of Teeth.	20	5.04	5.60	6.18	6.77	7.38
	30	7.99	8.61	9.24	9.89	10.52
	40	11.09	1.90	12.59	13.30	14.02
	50	14.74	15.48	16.23	17.00	17.77
	60	18.55	19.35	20.15	20.97	21.80
	70	22.65	23.50	24.36	25.24	26.12
	80	27.02	27.93	28.85	29.79	30.73
	90	31.69	32.66	33.63	34.62	35.63
	100	36.63	37.67	38.70	39.75	40.81
	120	47.40	48.54	49.69	50.85	52.03
	140	59.30	60.56	61.82	63.10	64.27
	160	72.35	73.73	75.10	76.39	77.90
	180	86.54	88.03	89.52	91.02	92.54
	200	101.88	103.48	104.98	106.70	108.34
	320	118.36	120.08	122.15	123.52	125.27

For weight of gear-wheels with number of teeth between figures given in left-hand column use weight given on horizontal line through nearest ten below the given number of teeth and under the figure in top line nearest last figure in number of teeth given; thus, 46 teeth = 13.30.

SHAFTING.

Shafting.—The formulæ and tables given below will be sufficient to enable the size of shaft required for any case likely to occur in the consideration of draw-spans to be readily determined. When the shaft is long and works through a limited number of revolutions the diameter should be large, in order that the angular deflection may not be excessive. The use of too small shafting has been one of the most common faults in draw-span design, and in many cases has led to the renewal of machinery that in other respects would have given satisfactory service.

A deflection of one degree in a length of twenty diameters is considered good practice in millwork, but for drawbridge machinery, if the shaft be long and there are many attachments to it, an angular deflection as great as this may cause the whole arrangement to work badly. The angular deflection for any twisting moment may be determined by the following

formulæ: A = the angular deflection in parts of one revolution, M = the twisting moment in foot-pounds, L = length of shaft in feet, d = diameter of shaft in inches; then for wrought iron $A = \frac{ML}{30000d^4}$, and for steel $A = \frac{ML}{36000d^4}$.

If the twisting moment M does not exceed $M = 50d^3$ for wrought iron and $M = 60d^3$ for steel, the angle of deflection will not exceed one degree for a length of shaft equal to 20 diameters. Thus if a 3-inch steel shaft have a twisting moment of $M = 60d^3 = 1620$ ft.-lbs., then

$$A = \frac{1620L}{36000 \times 81};$$

and if the length of shaft be 60 ft., then $A = 0.033$.

$360° \times 0.033 = 12°$. A deflection of one degree in 20 diameters = 12°.

Friction of Shaft-bearings.—For the slow motion of a hand-turning draw the friction of the shafts, if well oiled, would probably be about .025 of the pressure; but as the conditions of lubrication as well as the state of adjustment are uncertain, a coefficient of .06 has been used in the example considered. As the speed increases the coefficient will increase, and for higher speeds we may use

$$F = \frac{dl\sqrt{v}}{3.3}.$$

F = coefficient of friction, d = diameter of shaft, l = length of bearing, v = velocity in feet per second. It has been found that for loads up to 600 or 700 lbs. per square inch the friction depends upon the diameter, length of bearing, and velocity, and is independent of the pressure. With heavy loads and high speeds a coefficient of 0.11 should be used.

Collar Friction.—For the coefficient of friction on the collars, 0.06 to 0.1 (depending upon the method of oiling, etc.) should be used. This friction should be considered as

acting at the centre of gravity of the ring. For method of reducing friction of collar, where the thrust is heavy, see cut of ball-bearings.

*General Formulæ.**

$T = .196d^3s$ for round shafts; . . . . (*a*)

$T = .28d^3s$ for square shafts. . . . . (*b*)

$d =$ diameter of the shaft in inches;
$s =$ shearing strength in pounds per square inch;
$T =$ the torsional moment in inch-pounds; that is, the force in pounds multiplied by the length in inches of the lever through which the force acts, taking s at 40,000 and 50,000 lbs.; working value = 9000 and 11,200 lbs.

$T = 1760d^3$ for round iron shafts; . . . (*c*)

$T = 2200d^3$ for round steel shafts; . . . (*d*)

$T = 2520d^3$ for square iron shafts; . . . (*e*)

$T = 3150d^3$ for square steel shafts; . . . (*f*)

$d = \sqrt[3]{\frac{T}{1760}}$ for round iron shafts; . . . (*g*)

$d = \sqrt[3]{\frac{T}{2200}}$ for round steel shafts; . . . (*h*)

$d = \sqrt[3]{\frac{T}{2520}}$ for square iron shafts; . . . (*i*)

$d = \sqrt[3]{\frac{T}{3150}}$ for square steel shafts. . . . (*k*)

* Following tables on Strength of Shafting are from Pencoyd Pocket-book.

WORKING PROPORTIONS FOR CONTINUOUS SHAFTING, IRON OR STEEL.

No Bending Action except its Own Weight.

Diameter of Shaft in Inches.	Maximum Safe Torsional Moment in Inch-pounds.	Revolutions per Minute.					Minimum Distance in Feet between Bearings.
		100	150	200	250	300	
		H. P.	H. P.	H. P.	H. P.	H. P.	
$1\frac{1}{2}$	5,940	7	10	14	17	20	11.7
$1\frac{5}{8}$	7,552	9	13	17	21	26	12.4
$1\frac{3}{4}$	9,432	11	16	21	26	32	13.0
$1\frac{7}{8}$	11,602	13	20	26	33	40	13.6
2	14,080	16	24	32	40	48	14.2
$2\frac{1}{8}$	16,892	19	29	38	48	58	14.8
$2\frac{1}{4}$	20,048	23	34	46	57	68	15.4
$2\frac{3}{8}$	23,580	27	40	54	67	80	16.0
$2\frac{1}{2}$	27,500	31	47	63	78	94	16.5
$2\frac{3}{4}$	36,603	42	62	83	102	124	17.6
3	47,520	54	81	108	134	162	18.6
$3\frac{1}{4}$	60 417	69	103	137	172	206	19.7
$3\frac{1}{2}$	75,460	86	129	172	215	258	20.7
$3\frac{3}{4}$	92,812	105	158	211	264	316	21.6
4	112,640	128	192	256	320	384	22.6

WORKING PROPORTIONS FOR CONTINUOUS SHAFTING, IRON OR STEEL.

Transmitting Power and subject to Bending Action of Pulleys, Belting, etc.

Diameter of Shaft in Inches.	Maximum Safe Torsional Moment in Inch-pounds.	Revolutions per Minute.					Maximum Distance in Feet between Bearings.
		100	150	200	250	300	
		H. P.	H. P.	H. P.	H. P.	H. P.	
$1\frac{1}{2}$	5,940	5	7	10	12	14	6.8
$1\frac{5}{8}$	7,552	6	9	12	15	18	7.2
$1\frac{3}{4}$	9,432	8	11	15	18	22	7.5
$1\frac{7}{8}$	11,602	9	14	19	23	28	7.9
2	14,080	11	17	23	28	34	8.2
$2\frac{1}{8}$	16.892	14	21	27	34	42	8 6
$2\frac{1}{4}$	20,048	16	24	33	41	48	8.9
$2\frac{3}{8}$	23,580	19	29	38	48	58	9.2
$2\frac{1}{2}$	27,500	22	33	45	55	66	9.6
$2\frac{3}{4}$	36,603	24	36	48	60	72	10.2
3	47.520	39	58	77	96	116	10.8
$3\frac{1}{4}$	60,417	49	74	98	123	148	11.4
$3\frac{1}{2}$	75,460	61	92	123	153	184	12.0
$3\frac{3}{4}$	92,812	75	113	151	188	226	12.5
4	112,640	91	137	183	228	274	13.1

Shafts having Both Bending and Twisting.

$$T' = M + \sqrt{M^2 + T^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (l)$$

$M =$ bending moment in inch-pounds;

$T =$ twisting moments in inch-pounds;

$T' =$ a new twisting moment which, substituted for T in equations g to k, will give the desired proportions for the shaft.

Ratio of M to T.	Factor of Safety.	Divisor in Formulæ.	
		(g) for Iron.	(k) for Steel.
$M = .3T$ or less....................	$4\frac{1}{2}$	1760	2200
$M = .6T$ " "...................	5	1570	1960
$M = T$ " "...................	$5\frac{1}{2}$	1430	1790
$M =$ greater than T...............	6	1310	1640

Formulæ for Horse-power.

$V =$ revolutions per minute;

$HP =$ 396,000 inch-pounds per minute.

$$HP = \frac{6.28 \times T \times V}{396{,}000}, \quad T = \frac{63{,}057\ HP}{V}, \quad d = \sqrt[3]{\frac{36\ HP}{V}} \quad . \quad (o)$$

Deflection of Shafting.

$$l = \sqrt[3]{873d^2} \text{ for bare shafts; } . \quad . \quad . \quad . \quad . \quad . \quad . \quad (p)$$

$$l = \sqrt[3]{175d^2} \text{ for shafts carrying pulleys, etc.; } \quad . \quad (r)$$

which would be the maximum distance in feet between bearings for continuous shafting subjected to bending stress alone.

If the length is fixed and we desire the diameter of the shaft, we have

$$d = \sqrt{\frac{l^3}{873}} \text{ for bare shafting; } . \quad . \quad . \quad . \quad . \quad . \quad (s)$$

$$d = \sqrt{\frac{l^3}{175}} \text{ for shafting carrying pulleys, etc.} \quad . \quad (t)$$

Working Formulæ.

$$d = \sqrt[3]{\frac{50\, HP}{V}} \text{ for bare shafts;} \quad . \quad . \quad . \quad . \quad . \quad . \quad (u)$$

$$d = \sqrt[3]{\frac{70\, HP}{V}} \text{ for shafts carrying pulleys, etc.;} \quad (v)$$

$$l = \sqrt[3]{720 d^2} \text{ for bare shafts;} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (w)$$

$$l = \sqrt[3]{140 d^2} \text{ for shafts carrying pulleys, etc.} \quad . \quad . \quad (x)$$

Shafting-keys.

$$k = 0.16 \div \tfrac{1}{8}d; \qquad k' = 0.16 + \tfrac{1}{10}d.$$

Taper of key, .04 in. to .08 in. in 4 in.

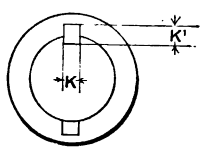


Fig. 33

Shaft	1/2"	5/8"	3/4"	1"	1½"	2"	2½"	3"	3½"	4"	4½"	5
□ **Key**	3/32"	1/8"	5/32"	7/32"	5/16"	7/16"	1/2"	9/16"	9/16"	5/8"	3/4"	7/8"

From Releaux.

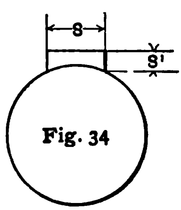


Fig. 34

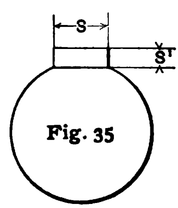


Fig. 35

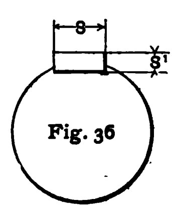


Fig. 36

If we call the diameter of the shaft D, the breadth of the key S, and the middle depth of the key S', we have:

For draft keys, $S = 0.24'' + \frac{D}{7}$; $S' = 0.16'' + \frac{D}{12}$.

For torsion keys, $S = 0.16'' + \frac{D}{5}$; $S' = 0.16'' + \frac{D}{10}$.

The taper of such keys is made about $\frac{1}{100}$.

For the more commonly occurring diameters we have the following proportions:

$D =$	1	2	3	4	5	6	7	8	9	10
					FOR DRAFT KEYS.					
$S =$	3/8″	1/2″	5/8″	13/16″	1″	$1\frac{1}{8}$″	$1\frac{1}{4}$″	$1\frac{3}{8}$″	$1\frac{1}{2}$″	$1\frac{5}{8}$″
$S' =$	1/4″	5/16″	7/16″	1/2″	9/16″	5/8″	3/4″	13/16″	7/8″	1″
					FOR TORSION KEYS.					
$S =$	3/8″	9/16″	3/4″	1″	$1\frac{3}{16}$″	$1\frac{3}{8}$″	$1\frac{9}{16}$″	$1\frac{3}{4}$″	2″	$2\frac{1}{8}$″
$S' =$	1/4″	3/8″	1/2″	9/16″	11/16″	3/4″	7/8″	1″	$1\frac{1}{16}$″	$1\frac{3}{16}$″

For shafts of less diameter than 1 in. we may make

$$S = \frac{D}{3}, \quad S' = \frac{D}{5}.$$

If several keys are used, they may be made the same dimensions as single keys. For hubs which have been forced on, and hence would be secure without any key, the dimensions for draft-keys may be used.

Bearings and Pivots, Springs, Cams, etc.

Bearings.—The bearings for shafts should be placed as near the points of loading as possible, and for low speeds and small loads the length of bearing should be once and one half to twice the diameter of the shaft. Where the load is heavy or speed great, the bearings are given a length of twice to four times the diameter. Where the bearing simply carries the weight of shaft, a length of once to once and one quarter the diameter is sufficient. Bearings of brass or a composition

of metals are used at important points. A bushing of Babbitt metal is found to give excellent results. The friction is low and the wearing properties of this metal are good. Two bearings made in this manner are shown in the cuts. As the speed increases, the length of the bearing should be increased about in the ratio given in table below.

N	=	100	150	200	250	400	750	1000
$l \div d$	=	1.25	1.5	1.75	2.0	2.5	3.5	4.0

N = number of revolutions per minute; l = the length of bearing in inches; d = diameter of shaft in inches.

Ample provision should be made for keeping the bearing well oiled, and all oil-holes should be easy of access. To aid in spreading the oil over the whole bearing-surface small grooves are often cut spirally around the bearing.

For thickness of metal and proportion of the various parts see cuts 39 and 40.

Load on Rollers.—*Seller's Centre.*—The rotating load per lineal inch on steel roller should not exceed that given by the following formula for steel rollers on steel plates:

$$P = 2625 \sqrt{d}.*$$

P = pressure per lineal inch of roller;

d = mean diameter of roller in inches.

Load on Wheels.—The load per lineal inch of face of wheel, while span is turning, should not exceed that given by the following formulæ, viz.:

$P = 705 \sqrt{d}$ for a cast-iron wheel on a cast-iron track;

$P = 900 \sqrt{d}$ " " " " " wrought-iron track.

For steel wheels use the following formulæ as to limit of pressure per lineal inch of wheel-face while the span is turning, viz.:

$P = 1905 \sqrt{d}$ for a steel wheel on a cast-iron track;

$P = 1515 \sqrt{d}$ " " " " wrought-iron track;

$P = 1750 \sqrt{d}$ " " " " steel track.

* It is often specified that the load shall not exceed $P = 1750 \sqrt{d}$.

In which formulæ

$P =$ allowed pressure per lineal inch of face of wheel;
$d =$ diameter of wheel in inches.

Pivots.

FORMULÆ FOR PIVOTS.

Wrought Iron or Steel on Bronze.

Slow-moving pivots $\begin{cases} p = 1422. \\ d = 0.035\sqrt{P}. \end{cases}$

$n =$ or < 150 $\begin{cases} p = 700. \\ d = 0.05\sqrt{P}. \end{cases}$

$n > 150$ $\begin{cases} a = 75. \\ d = 0.004\sqrt{Pn}. \end{cases}$

Cast Iron on Bronze.

Slow-moving pivots $\begin{cases} p = 700. \\ d = 0.05\sqrt{P}. \end{cases}$

$n =$ or < 150 $\begin{cases} p = 3\ 50. \\ d = 0.07\sqrt{P}. \end{cases}$

$n > 150$ $\begin{cases} a = 75 \\ d = 0.006\sqrt{Pn}. \end{cases}$

Iron or Steel on Lignum Vitæ.

Slow-moving pivots $\begin{cases} p = 2844 \\ d = 0.017\sqrt{P}. \end{cases}$

$n =$ or < 150 $\begin{cases} p = 1422. \\ d = 0.035\sqrt{P}. \end{cases}$

$n > 150$ $\begin{cases} p = 1422. \\ d = 0.035\sqrt{P}. \end{cases}$

TABLE OF SAFE LOAD FOR STEEL ON BRONZE.

d.	$0.035\sqrt{P}$ Slow. Load.	$0.05\sqrt{P}$. Under $150R$. Load.	$0.07\sqrt{P}$. Over $150R$. Load.
1	816	398	204
1.25	1,275	622	319
1.50	1,836	895	459
1.75	2,500	1,219	625
2.00	3,265	1,592	816
2.25	4,132	2,016	1,033
2.50	5,102	2,488	1,275
2.75	6,173	3,011	1,543
3.00	7,347	3,494	1,836
3.25	8,622	4,205	2,155
3.50	10,000	4,877	2,500
3.75	11,479	5,599	2,869
4.00	13,061	6,370	3,265
4.25	14,745	7,192	3,686
4.50	16 530	8,063	4,132
4.75	18,418	8,983	4,604
5.00	20,498	9,954	5,102
5.25	22,140	10,974	5,535
5.50	24,694	12,044	6,673
5.75	26,990	13,164	6,747
6.00	29,388	14,334	7,344
6.25	31,890	15,630	7,972
6.50	34,490	16,900	8,623
6.75	37,190	18,220	9,298
7.00	41,690	19,600	10,000

The above table is made from the formula $P = 816d^2$ for slow speeds, and $P = 816d^2\frac{a}{n}$ for high speeds. For cast iron on bronze use one half the above values and for steel or iron on lignum vitæ use double the values given in the table. $n =$ the number of revolutions per minute, $p =$ the pressure per square inch, $P =$ total pressure, $d =$ diameter of pivot, and the constant $a = 75$.

Formulas for Springs.

By George R. Henderson, Mechanical Engineer, N. & W. R. R

For Elliptic Springs. — $P =$ maximum static load in pounds; $S =$ corresponding fibre-strain in leaves taken at 80,000 lbs.; $N =$ number of leaves (in full elliptic), half the total leaves; $B =$ width of leaves in inches $H =$ thickness of leaves in inches; $L =$ span (or length) of spring in inches when loaded; $F =$ deflection of spring under load P in inches; $E =$ modulus of elasticity taken at 30,000,000. Then

$$P = \frac{2SNBH^2}{3L}, \text{ and reducing } P = \frac{53333NBH^2}{L}.$$

For half elliptic $F = \frac{55PL^3}{16ENBH^3}$, and reducing $F = .000611\frac{L^3}{H}$.

For full elliptic $F = \frac{12PL^3}{16ENBH^3}$, and reducing $F = .00133\frac{L^3}{H}$.

For Helical Springs.—$P =$ load when spring is down solid, in pounds; $S =$ maximum shearing fibre-strain in bar taken at 80,000; $D =$ diameter of steel in inches; $R =$ radius of centre of coil in inches; $L =$ length of bar before coiling in inches; $G =$ modulus of shearing elasticity taken at 12,600,000; $F =$ deflection of spring under load, in inches; $H =$ height of spring free in inches; $h =$ height of spring solid in inches; $\pi = 3.1416$. Then

$$P = \frac{S\pi D^3}{16R};\ F = \frac{32PR^2L}{G\pi D^4};\ H = \frac{LD}{2\pi R};\ H = h + F;$$

and substituting proper constant,

$$F = .08\frac{R^2H}{D^2};\ H = h(1 + .08\frac{R^2}{D^2});\ P = 15.714\frac{D^3}{R}.$$

The most generally preferred ratio for size is $D = 5d$, where $D =$ outside diameter of coil. It is customary to make the static load about one half the solid load.

Helical Springs.

By D. K. Clark.

$$E = \frac{d^3 \times w}{D^4 \times C}; \quad \text{. . . . .} \quad (1)$$

$$D = \sqrt[3]{\frac{w \times d}{3}} \text{ for round steel;} \quad \text{. . .} \quad (2)$$

$$D = \sqrt[3]{\frac{w \times d}{4.29}} \text{ for square steel.} \quad \text{. . .} \quad (3)$$

$E =$ compression or extension of one coil, in inches; $d =$ diameter from centre to centre of steel bar composing the spring, in inches; $w =$ the weight applied, in pounds; $D =$ the diameter, or the side of square, of the steel bar of which the spring is made, in sixteenths of an inch; $C =$ a constant which, from experiments made, may be taken as 22 for round steel and 30 for square steel.

Eccentrics.

Eccentrics.—An eccentric is nothing more than a crank in which (if the crank-arm is R and the shaft diameter D) the crank-pin diameter d' is made so great that it exceeds $D + 2R$, or is greater than the shaft and twice the throw. The simpler forms of eccentric construction are shown in the illustrations. The most practical of these is that shown in Fig. 37*b*, the flanges on the strap, as shown in the section, serving to retain the oil and insure good lubrication.

The breadth of the eccentric is $1\frac{1}{2}d$ to $3d$, the same as that of the equivalent overhung journal subjected to the same pressure. For the depth of flange a we have

$$a = 1.5e = 0.07l + 0.2$$

From which the other dimensions can be determined as in the illustrations

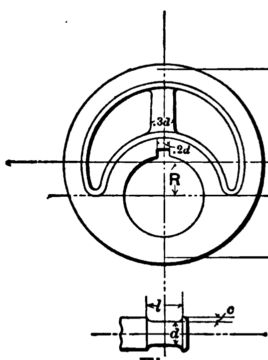


Fig. 37

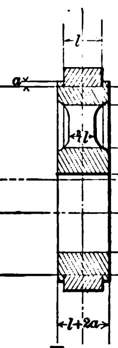


Fig. 37ᵃ

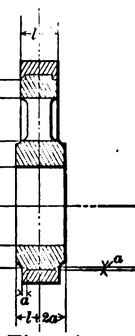


Fig. 37ᵇ

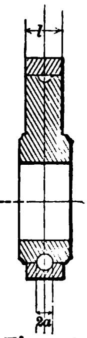


Fig. 37ᶜ

Hooks.

Formulas prepared by the YALE & TOWNE MANUFACTURING Co.

Δ = capacity of hook in tons of 2000 lbs.

$D = .5\Delta + 1.25$	$G = .75D;$
$E = .64\Delta + 1.60$	$O = .363\Delta + .66;$
$F = .33\Delta + .85;$	$Q = .64\Delta + 1.60;$
$H = 1.08A;$	$L = 1.05A;$
$I = 1.33A;$	$M = .50A;$
$J = 1.20A;$	$N = .85B - 16$
$K = 1.13A;$	$U = .866A.$

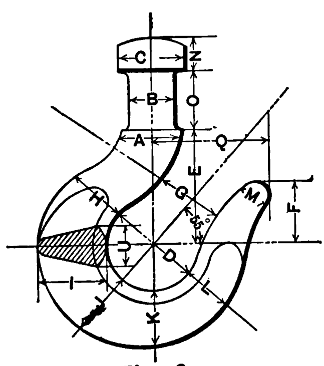


Fig. 38

Capacity of hook......	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	1	$1\frac{1}{2}$	2	3	4	5	6	8	10	tons.
Dimension A.........	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{7}{8}$	$3\frac{1}{4}$	in.

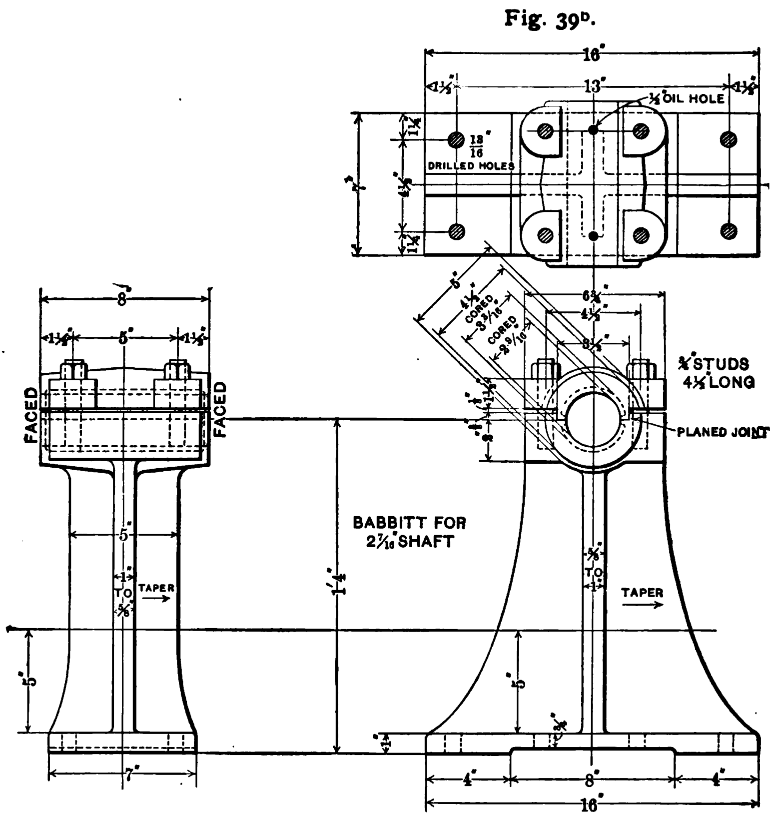


Fig. 39. Fig. 39a.

SHAFT-BEARING.

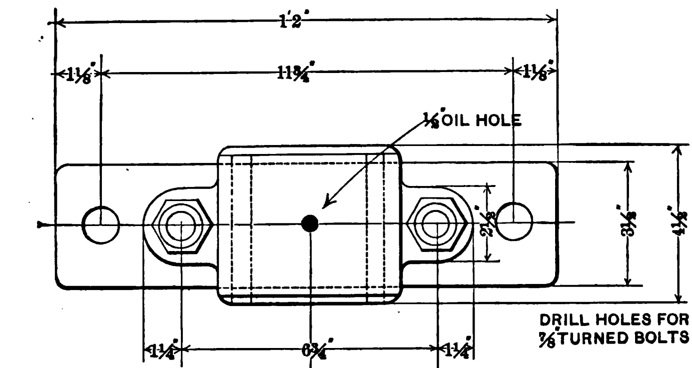


Fig. 40a

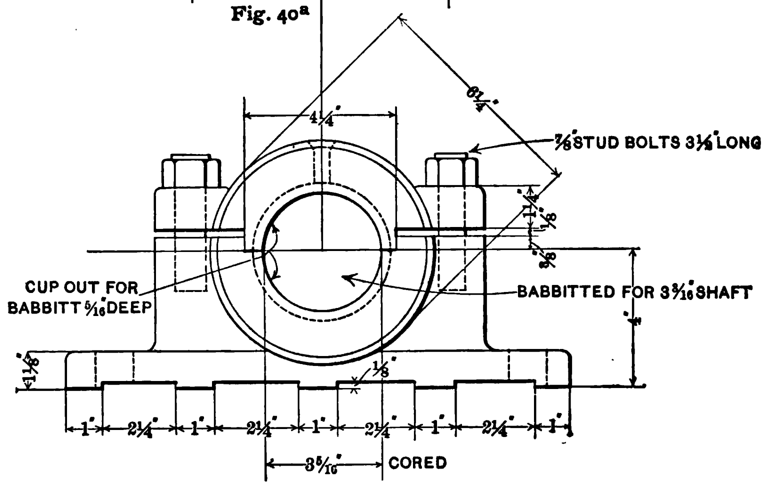
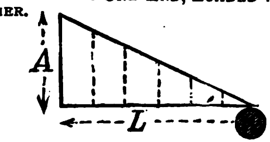
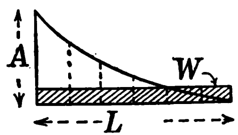
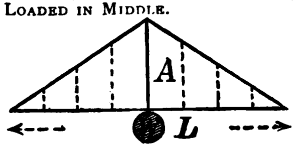
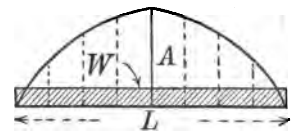
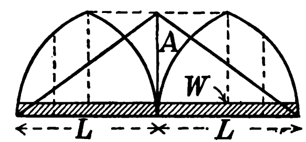


Fig. 40.

SHAFT-BEARING.

BENDING MOMENTS AND DEFLECTIONS FOR BEAMS OF UNIFORM SECTION.

Form of Beam and Position of Load.	Max. Bending Moment.	Max. Shearing Stress.	Deflection.
(1) BEAM FIXED AT ONE END, LOADED AT OTHER.  Draw triangle having $A = W.L.$ Vertical lines give bending moments at corresponding points on beam.	At point of support $= WL.$	At point of support $= W.$	At end of beam$=\frac{WL^3}{3EI}.$
(2) BEAM FIXED AT ONE END, LOAD UNIFORMLY DISTRIBUTED.  Draw parabola having $A = \frac{W.L}{2}.$ Ordinates give bending moments at corresponding points on beam.	At point of support $= \frac{WL}{2}.$	At point of support $= W.$	At end of beam $= \frac{WL^3}{8EI}.$
(3) BEAM SUPPORTED AT BOTH ENDS, LOADED IN MIDDLE.  Draw triangle having $A = \frac{W.L}{4}.$ Vertical lines give bending moments at corresponding points on beam.	At middle of beam $= \frac{WL}{4}.$	At point of support $= \frac{W}{2}.$	At middle of beam $= \frac{WL^3}{48EI}.$
(4) BEAM SUPPORTED AT BOTH ENDS, LOAD UNIFORMLY DISTRIBUTED.  Draw parabola having $A = \frac{WL}{8}.$ Ordinates give bending moment at corresponding points on beam.	At middle of beam $= \frac{WL}{8}.$	At point of support $= \frac{W}{2}.$	At middle of beam $= \frac{WL^3}{76.8EI}$
(5) CONTINUOUS BEAM ON THREE SUPPORTS. UNIFORM LOAD.  Draw two parabolas having $A = \frac{WL}{8}.$ Ordinates give bending moments.	At middle of support $= \frac{WL}{8}.$	At middle support $= \frac{5}{8}W \times 2.$	At centre of span $= \frac{WL^3}{31.9EI}.$

NOTE.—In case Fig. 45, W = total load on one arm of beam.

W = total load;
L = length of beam;
E = modulus of elasticity;
I = moment of inertia.

DRAW-SPAN MOMENTS AND SHEARS.

(See Fig. 11.)

COEFFICIENTS C' FOR LOADS IN FIRST ARM AND COEFFICIENTS C^2 AND D^2 FOR LOADS IN SECOND ARM.

Number or Panels in Half-span.	*B* *B'*	*C* *C'*	*D* *D'*	*E* *E'*	*F* *F'*	*G* *G'*	*H* *H'*	*I* *I'*	Totals.
4	.0586	.0938	.0820						.2344
5	.048	.084	.096	.072					.300
6	.0406	.0740	.0937	.0925	.0637				.3645
7	.0350	.0656	.0875	.0962	.0875	.0568			.4285
8	.0308	.0586	.0806	.0938	.0952	.0820	.0513		.4923
9	.0274	.0527	.0740	.0891	0960	.0925	.0767	.0466	.5550

COEFFICIENTS D' FOR LOADS IN FIRST ARM.

4	.691	.406	.168						1.265
5	.752	.516	.304	.128					1.700
6	.792	.592	.406	.241	.103				2.134
7	.822	.649	.484	.332	.198	.086			2.571
8	.844	.691	.544	.406	.280	.168	.074		3.007
9	.861	.725	.592	.466	.348	.241	.146	.065	3.444

VALUES OF E' FOR LOADS IN FIRST ARM.

4	.810	.842	.900						
5	.807	.827	.862	.916					
6	.805	.818	.842	.879	.929				
7	.803	.813	.830	.856	.893	.943			
8	.803	.810	.824	.842	.868	.900	.943		
9	.801	.809	.820	.832	.852	.879	.910	.950	

LOADS FOR MAXIMUM NEGATIVE MOMENTS—FIRST ARM.

4	—	—	—	—	—	—	—	—	—
5	—	—	—	—	—	—	—	—	—
6	*B*	*C*					For maximum at *F*		
7	*B*	*C*	*D*	*E*			" " " *G*		
8	*B*	*C*	*D*	*E*	*F*		" " " *H*		
9	*B*	*C*	*D*	*E*	*F*	*G*	" " " *I*		

All loads on second arm in each case. All loads cause negative moments over pier.

LOADS FOR MAXIMUM POSITIVE MOMENTS—FIRST ARM.

									Max. at
4	*B*	*C*	*D*						*B* to *D*
5	*B*	*C*	*D*	*E*					*B* to *E*
6	*B*	*C*	*D*	*E*	*F*				*B* to *E*
6			*D*	*E*	*F*				*F*
7	*B*	*C*	*D*	*E*	*F*	*G*			*B* to *F*
7					*F*	*G*			*G*
8	*B*	*C*	*D*	*E*	*F*	*G*	*H*		*B* to *G*
8						*G*	*H*		*H*
9	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*B* to *H*
9							*H*	*I*	*I*

SHEARS: All loads on second arm cause negative shear in first arm.
Loads moving A towards Z cause negative shear in first arm
Loads moving Z towards A cause positive shear in first arm.

P_1 = any load in first arm.
P_2 = any load in second arm.
S_1 = reaction at A from P_1 or P_2.
M_2 = moment at pier from P or P_2.
X_0 = distance from A to point of zero moment in first arm.
L = length of half-span.

$M_2 = C'P_1L$ or C^2P_2L.
$S_1 = D'P_1$ or D^2P_2.
$X_0 = E'L$.

WEB-STRESSES: Max. stress in any $\left\{\begin{matrix}\text{mem.,}\\\text{web,}\end{matrix}\right\}$ load moving A to Z, is when load extends from A to piece in question.

Max. stress in any $\left\{\begin{matrix}\text{mem.,}\\\text{web,}\end{matrix}\right\}$ load moving Z to A, is when load extends from Z to piece in question.

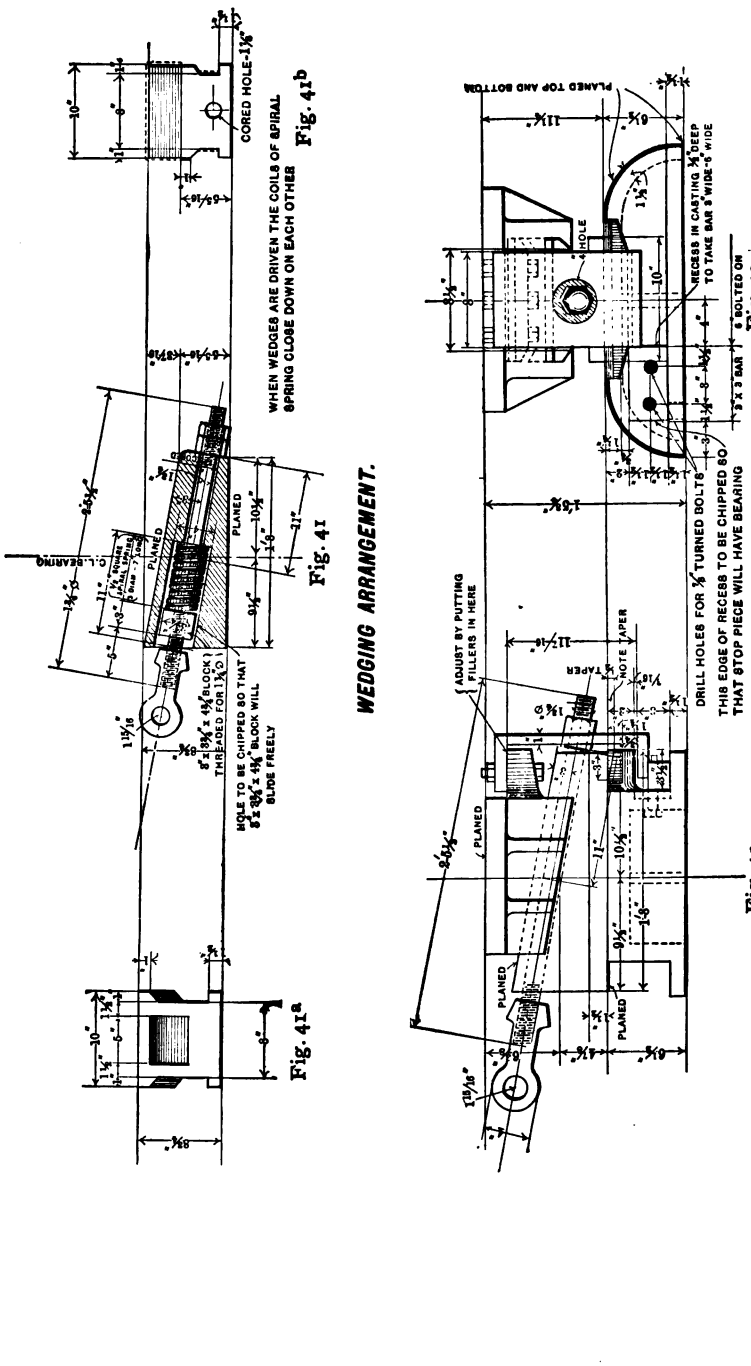


Fig. 41ᵇ

Fig. 41

Fig. 41ª

WEDGING ARRANGEMENT.

Fig. 43

Fig. 42

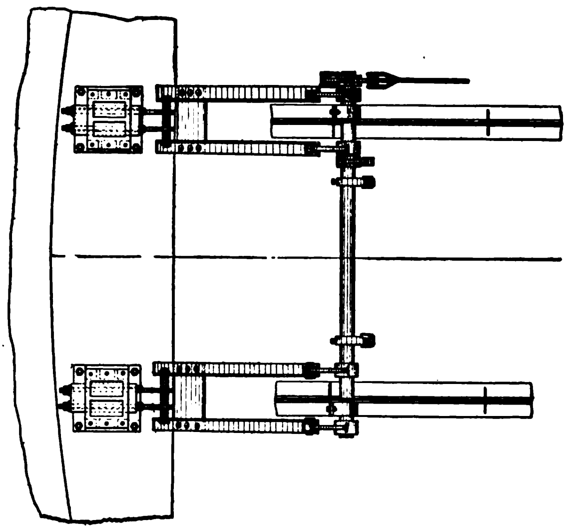


Fig. 44.

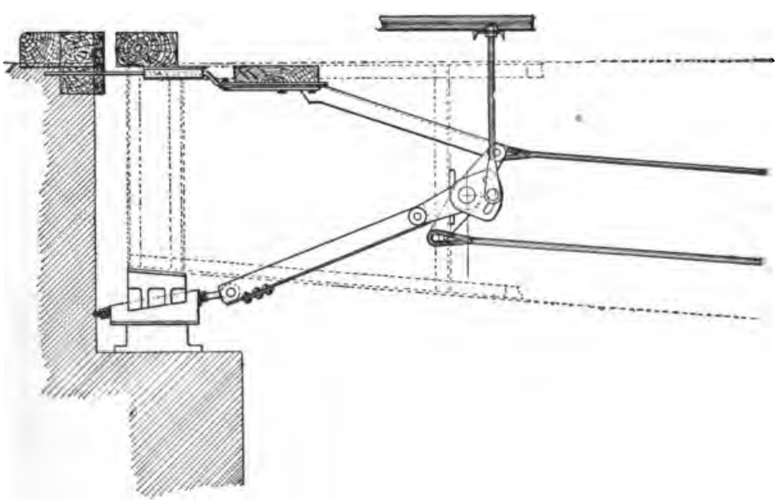


Fig. 45.

END MACHINERY.

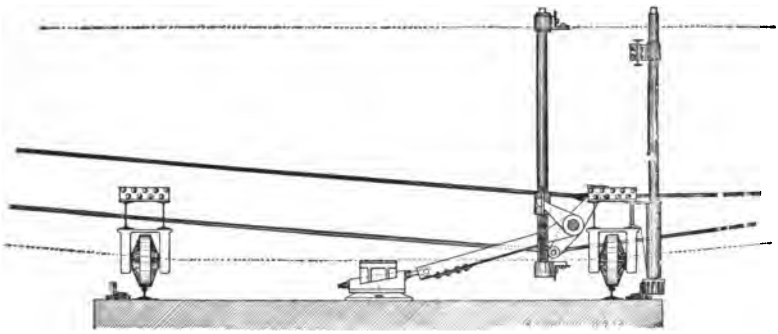


Fig. 46.

CENTRE MACHINERY.

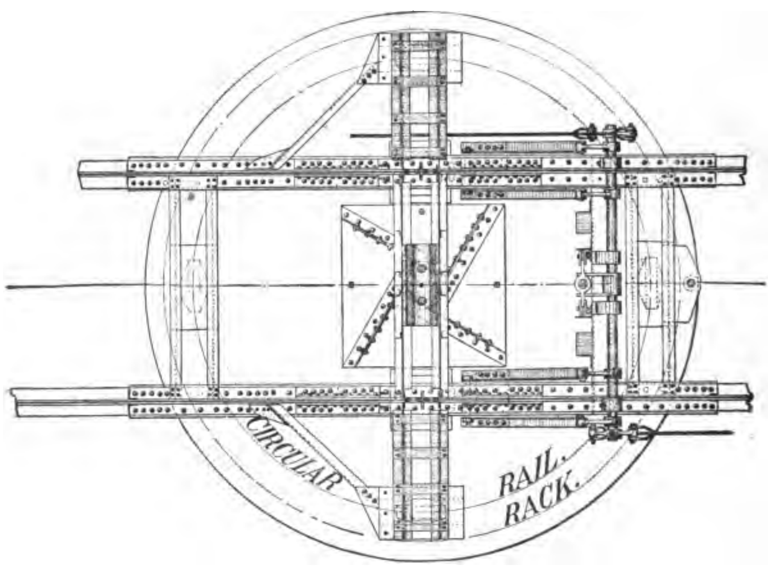


Fig. 47.

CENTRE MACHINERY

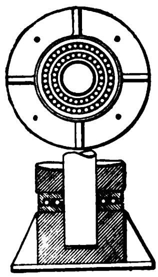


Fig. 48.
BALL-BEARING CENTRE.

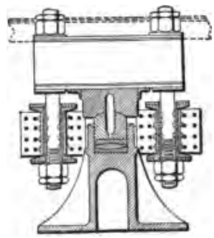


Fig. 49.
PIVOT CENTRE.

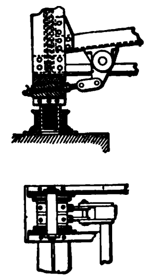


Fig. 50.
ADJUSTABLE END WEDGE.

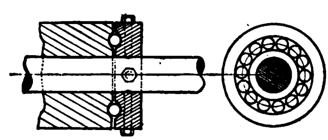


Fig. 51.
SHAFT BALL-BEARING.

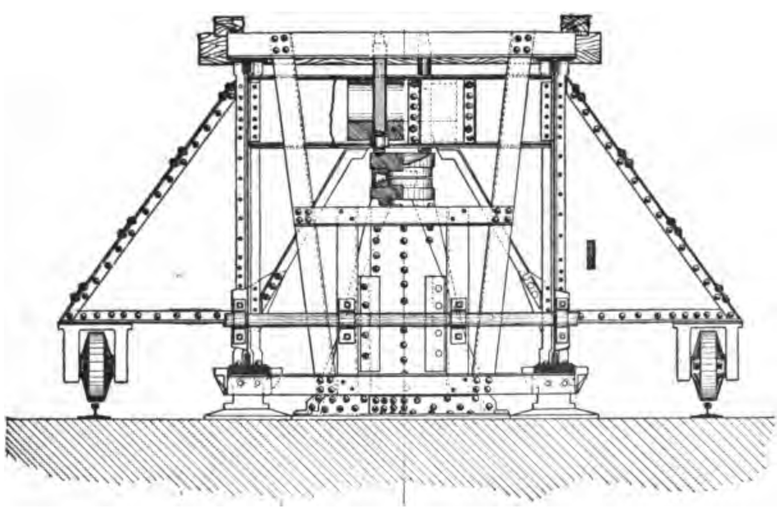


Fig. 52.
CENTRE ON CONICAL ROLLERS.

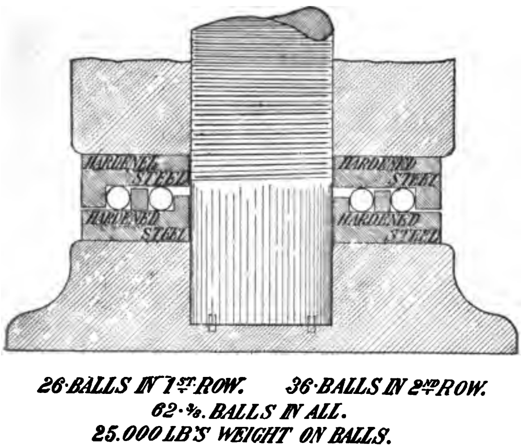


Fig. 53.

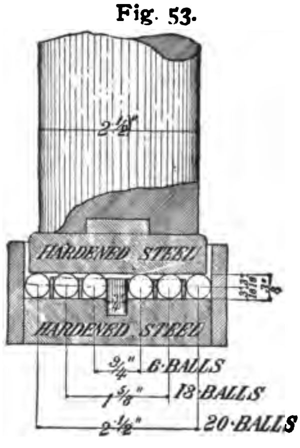


Fig. 54.

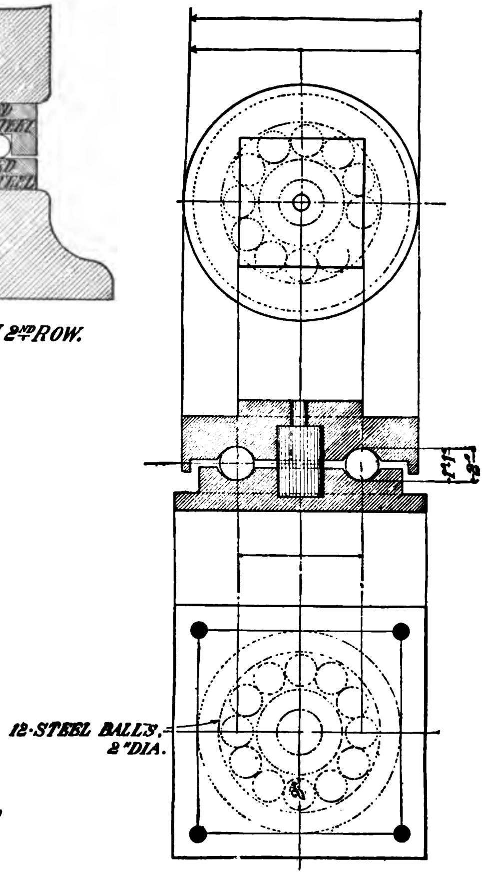


Fig. 55.

BALL-BEARINGS.

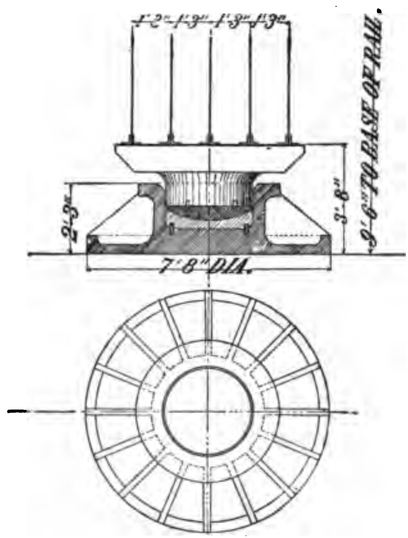


Fig. 56.
CENTRE PIVOT.

Pivot 33″ diam. to be forged in steel. Friction disks turned and ground spherically to a 36″ radius. Upper part steel. Lower part phosphor-bronze. Base of cast iron, to be faced top and bottom, turned inside.

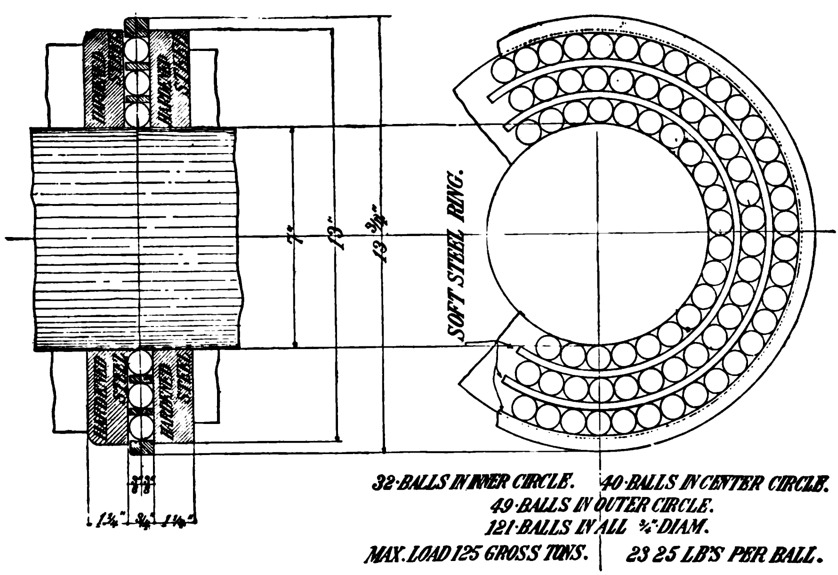


Fig. 57.
BALL-BEARING.

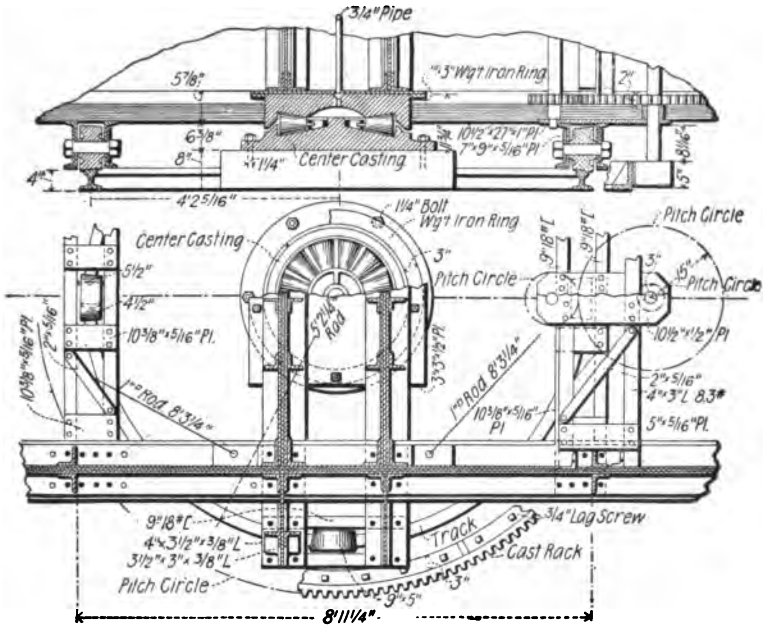


Fig. 58.

CENTRE FOR SMALL DRAW.

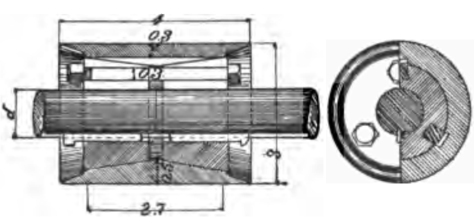
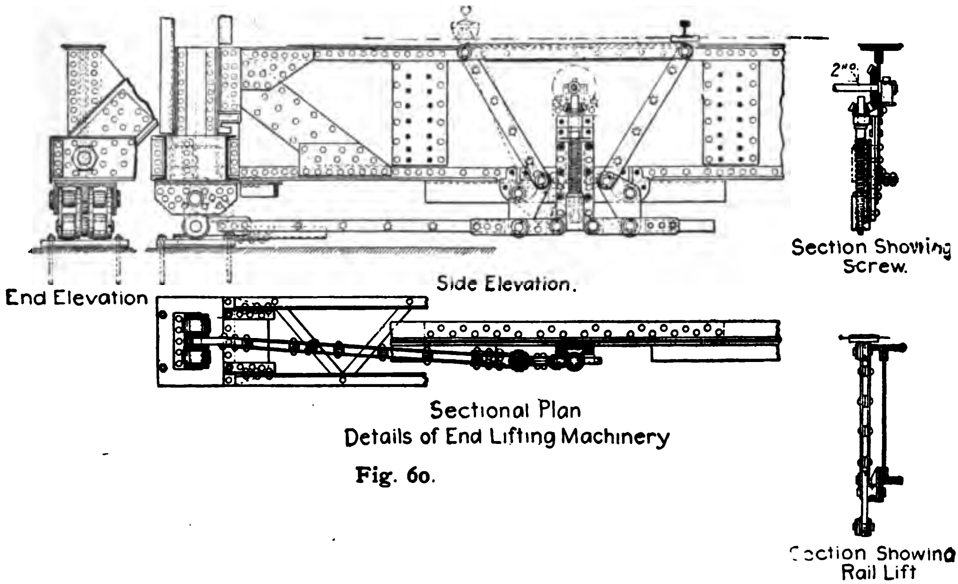


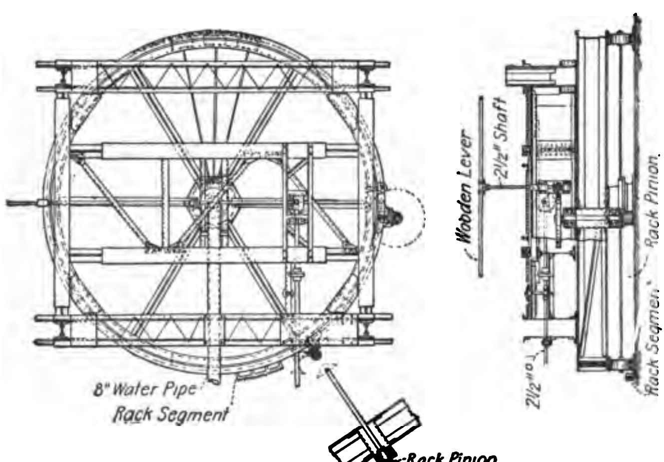
Fig. 59

SELLERS COUPLING FOR SHAFTING.



Details of End Lifting Machinery

Fig. 60.



Details of Turn Table.

Fig. 61.

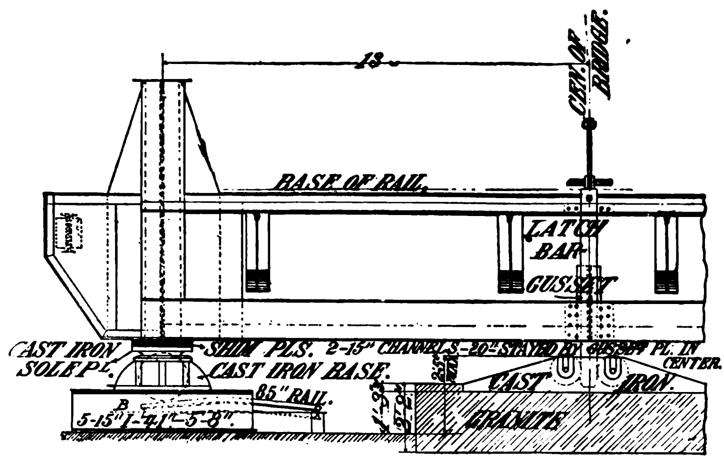


Fig. 62.
END SUPPORTS, LATCH, ETC.

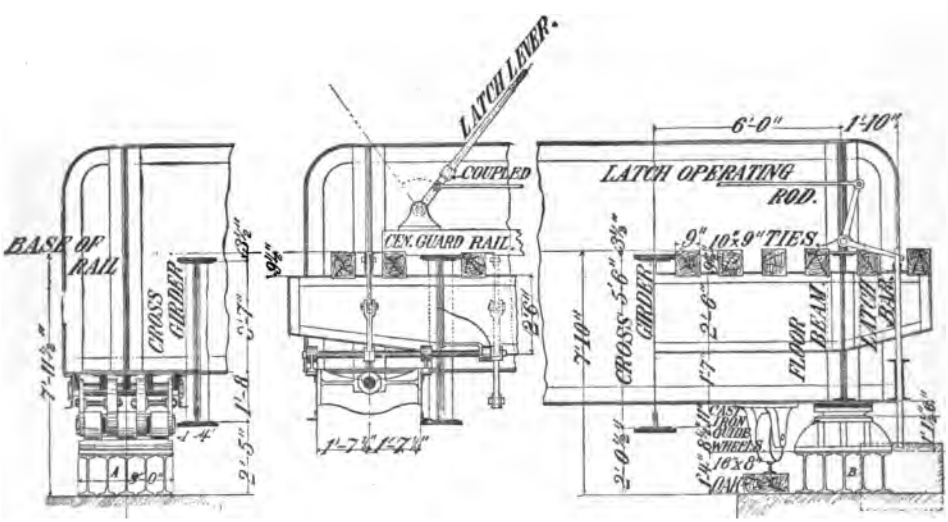


Fig. 63.
END LIFT, LATCH MACHINERY, ETC.

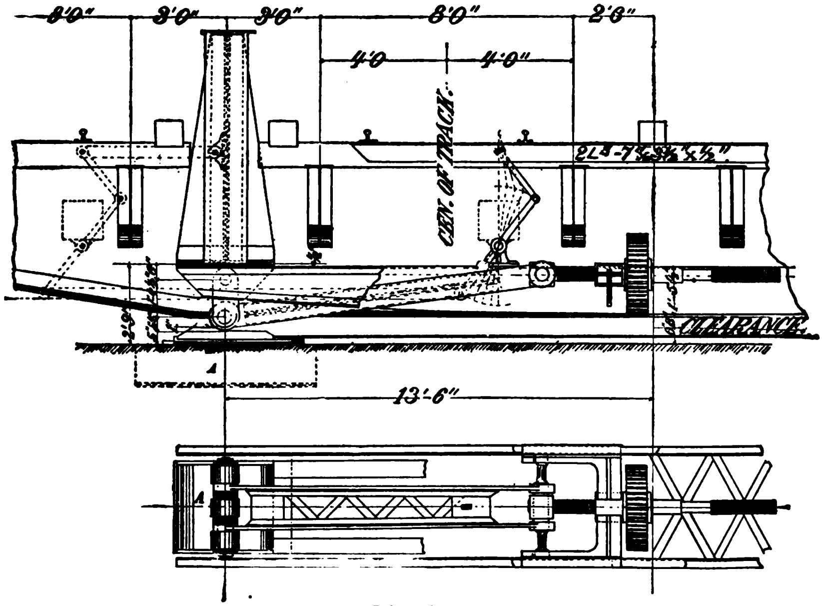


Fig. 64.

END LIFT.

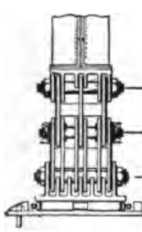


Fig. 65b.

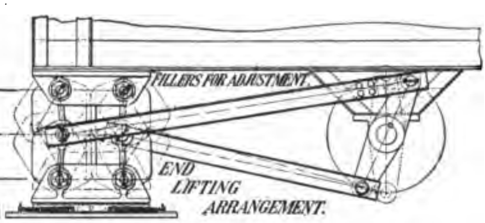


Fig. 65.

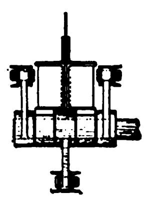


Fig. 65a.

END LIFT.

When draw is closed and ends are raised the middle pins of toggle-joint stand ½" inside of vertical line through top and bottom pins to prevent the toggle from opening In above position castings bear against each other.

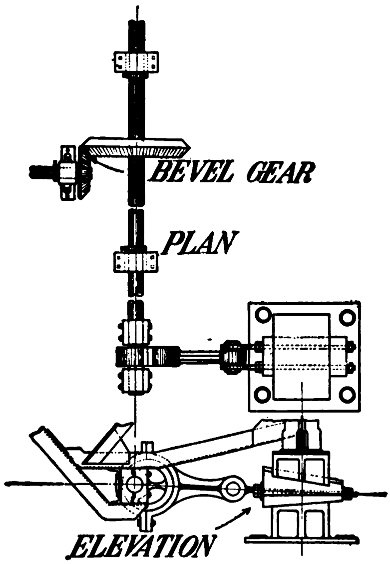


Fig. 66.

WEDGING GEAR.

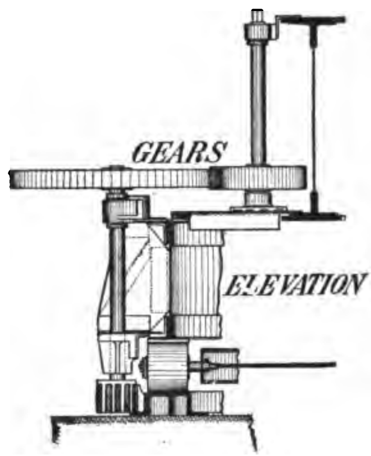


Fig. 67.

TURNING GEAR.

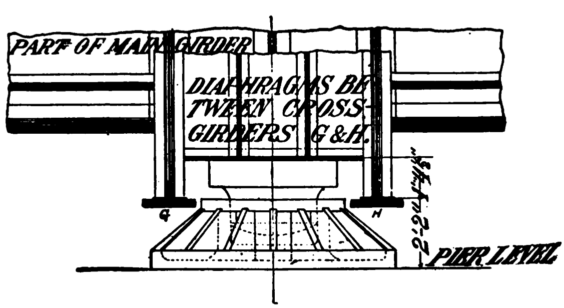


Fig. 68.

PIVOT CENTRE.

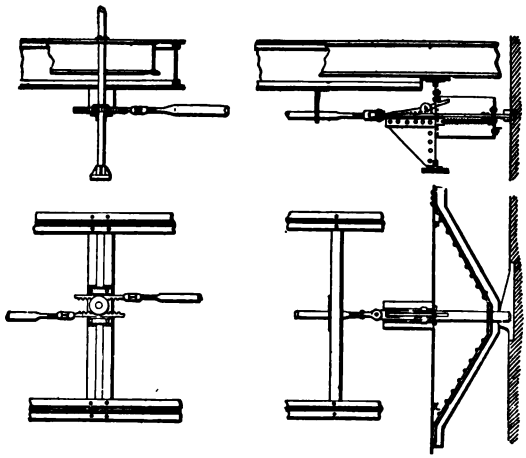


Fig. 69.

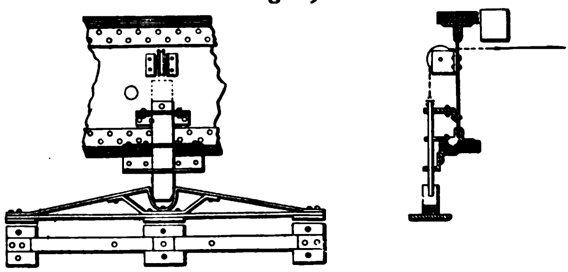


Fig. 70.
LATCHING DEVICES.

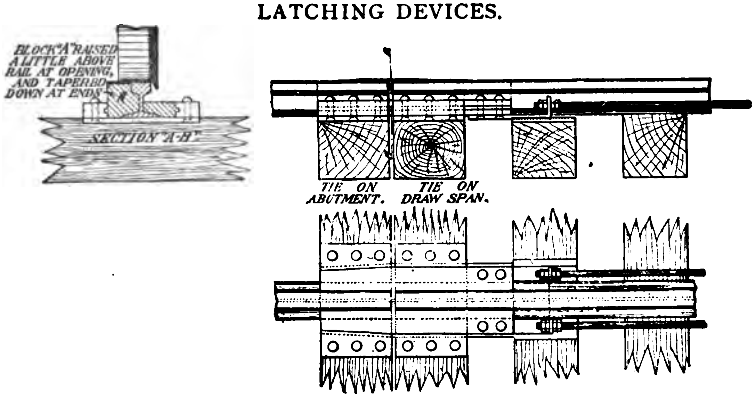


Fig. 71.
SLEEVE FOR CLAMPING RAILS.

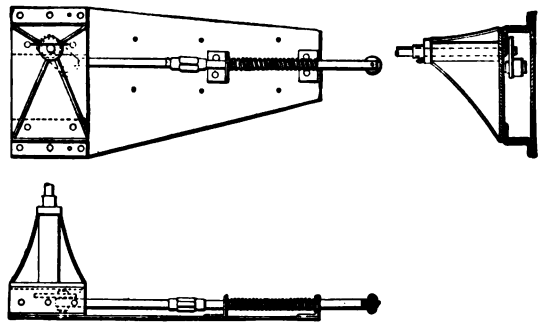


Fig. 72.

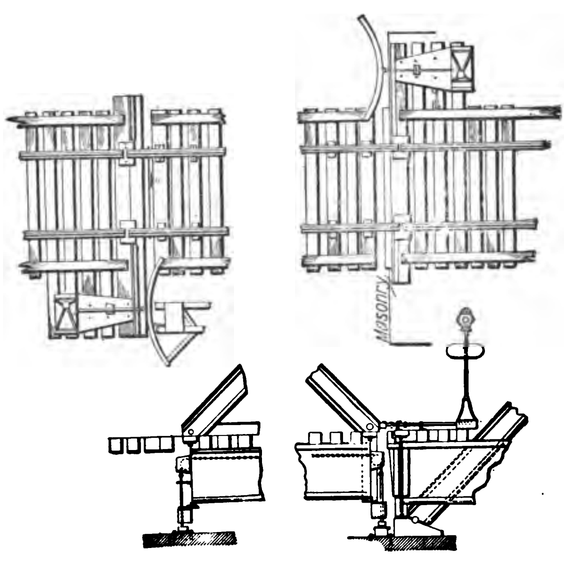


Fig. 73.

MACHINERY FOR OPERATING SAFETY-SIGNALS.

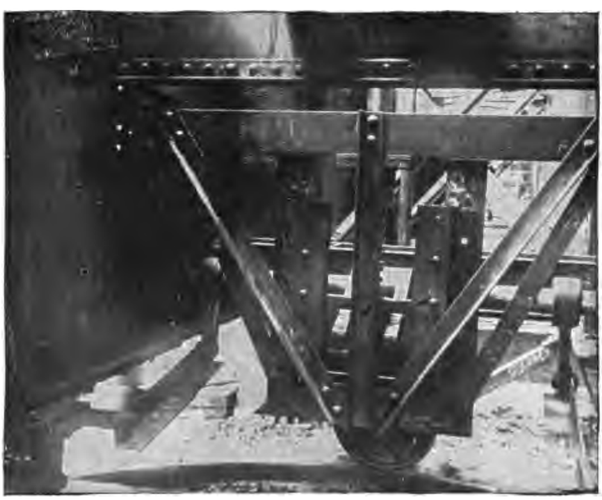
VIEWS SHOWING PLATE-GIRDER DRAW IN PROCESS OF CONSTRUCTION.

Balance-wheel and Centre Wedge.

End Wedges and Portion of Machinery in Position.

VIEWS SHOWING PLATE-GIRDER DRAW IN PROCESS OF CONSTRUCTION.

End Wedging Arrangement.



Portion of Machinery at Centre.

EXPLANATORY NOTES.

Where the term "moment of resistance" and the letter *R* designating the same have been employed in this work, they are used as indicating the moment of resistance for a fibre-stress of 1; or the term indicates the "section modulus" as given by some authors.

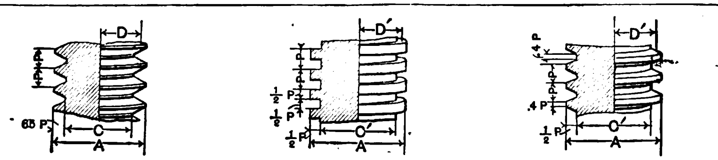
2. In Case 5, page 68, for continuous beams on three supports, note that the moments are obtained by scaling the ordinates between the curve and the inclined line, and not by scaling between the curve and the horizontal line as in the other cases.

3. On page 16 it will be noticed that the centre moments have been given for the loads on one arm only. The moments for the loads on the other arm are the same, and have been included in obtaining the total moment.

Friction of Worm-thread. (See page 38.)—The efficiency of the worm is very much reduced by the friction. In many cases a coefficient as high as 0.15 would be nearer correct than 0.10. The formula for the available vertical force is $W = \dfrac{r(F + F_1)}{\dfrac{P}{6.28} + cD}$, where W = vertical force, r = radius of turning lever, F = force at end of turning lever to overcome the vertical force W, F_1 = force at end of turning lever to overcome the friction produced by W, P = pitch of the worm-thread, D = the distance from centre of shaft to the centre of the worm-thread, c = the coefficient of friction. A force of 1 lb. at the end of a 6-ft. lever gives an available vertical force on the worm-nut, after deducting the friction of the thread and of the guides, as follows:

Diameter of Shaft.	Pitch.	Size of Thread.	W, in Pounds.
3½″	1½″	¾ in. sq.	161
3¼″	1¼″	⅝ "	182
3″	1″	½ "	207
2¾″	¾″	⅜ "	242

WORKING VALUES FOR WORM-SHAFTS.



V THREAD — SQUARE THREAD — EMERY THREAD

A	B	P	C	Area of A.	Area of C.	Safe Tensile Strain Iron at 10,000.	Safe Tensile Strain Steel at 12,500.	D	D'	W	W'
1½	6	.167	1.284	1.767	1.230	12,300	15,375	.696	.708	91.7	90.5
1⅝	5½	.182	1.389	2.073	1.496	14,960	18,700	.754	.767	84.5	83.2
1¾	5	.200	1.491	2.405	1.750	17,500	21,875	.810	.825	78.3	77.1
1⅞	5	.200	1.616	2.761	2.000	20,000	25,000	.873	.885	73.7	72.7
2	4½	222	1.712	3.141	2.300	23,000	28,750	.928	.944	68.8	67.8
2¼	4½	.222	1.962	3.976	2 990	29,900	37,375	1.053	1.070	62.1	61.3
2½	4	.250	2.176	4.908	3.640	36,400	45,500	1.169	1.188	55.8	55.0
2¾	4	.250	2.426	5.939	4.806	48,060	60,070	1.294	1.313	51.3	50.7
3	3½	.286	2.629	7 068	5.411	54,110	67,638	1.407	1.429	46.8	46.2
3¼	3½	.286	2.879	8.295	6.491	64,910	81,137	1.530	1.554	43.6	42.6
3½	3¼	.308	3.100	9.621	7.080	70,800	88.500	1.650	1.673	40 5	40.0
3¾	3	.333	3.317	11.044	8.395	83,950	104,930	1.767	1.792	37.7	37.3
4	3	.333	3.567	12.566	9.970	99,700	124,620	1 892	1.917	35.6	35.2
4¼	2⅞	348	3 798	14.186	11.144	111,440	139,300	2.012	2.038	33.5	33.2
4½	2¾	364	4.028	15.904	12.567	125,670	157,080	2.132	2.159	31 8	31.4

Number of threads per inch on above bolts is the number given in the Sellers System.

A = external diameter; B = number of threads per inch; C = diameter at root of thread; D, D' = radius of centre of thread; W (for v thread), W' (for square thread) = the weight which can be raised by a force of 1 lb. with a leverage of 1 foot. Coefficient of friction = .15.

DESIGNING OF DRAW-SPANS.

PART SECOND.

DRAW-SPANS.

THERE has been a very great departure in the last few years from the swing-bridge, which for a long time was practically the only type of draw in use in this country. Conditions similar to those which exist in Chicago have compelled engineers to adopt other forms. Property became too valuable to permit the use of the space required for the operation of the swing-span. Chicago offers more examples than any other city of the plans, more or less successful, which have been devised and tried to meet the new requirements. It is interesting to note that the method at present received with the greatest favor by engineers is merely a perfected and enlarged example of a type of draw in use hundreds of years ago. The bascule was probably the earliest form of a movable bridge.

The several types now in use are the swing-span, the pontoon, the vertical lift, the bascule, of which the rolling lift is one form, the folding lift, the folding swing, and the rolling draw, which moves back on the abutments in a line parallel to itself when closed.

Swing-span.—The swing is still the form of draw most used, and except in crowded cities or under special conditions it is the simplest, cheapest, and best. The tendency at the

present time is to combine with this form of draw such machinery or such construction as will make the bridge when closed act as two independent fixed spans. One method is to lift the whole span, when it is to be turned, from its bearings, to allow it to swing clear. Several of the latest draws built in England have hydraulic jacks for this purpose. Another plan is to shorten the top chord by means of cams or levers, which causes the ends to rise sufficiently to clear when swinging. When some device of this kind is used a bridge is obtained which when closed has all the advantages of a fixed span.

The Pontoon.—The pontoon draw can only be used where the distance from bridge to water-level is small. On a stream subject to sudden floods it would prove very unsatisfactory. The time required to open would be much greater than with the other types, and it is hardly likely this style of draw will be much used except possibly where the conditions are unusually favorable.

The Vertical Lift.—The vertical lift has some great advantages (see Halsted St. Draw, page 219). Its disadvantages are heavy first cost and maintenance, expensive operation, and the great height to which it must be lifted each time. In appearance it has decided advantage over all the other types, and may be made a very attractive structure. The towers offer opportunity for effective design.

The Bascule.—Some form of the bascule is the favorite with engineers at the present time. Several varieties are shown in the cuts. This type seems the best adapted to meet the requirements which exclude the use of the swing-span. It is quickly opened and closed. When open it affords a sure barrier against accidents from teams or foot-passengers falling into the opening. It is more ornamental than the swing-span, and it seems possible to build it in lengths sufficient for most cases where such a draw would be required. Some very ingenious methods of counterweighting have been devised,

the force exerted by the counterweight being just sufficient to balance the weight of the span at all points of the lift, the force varying from a maximum at the beginning of the lift to nearly zero at the end.

The Folding Lift.—The Weed St. and Canal St. draws in Chicago are examples of the folding-lift bridge. While this form of draw is cheaper than the bascule proper, it would seem to be limited to use in short highway spans. The floor being in four parts connected by hinges, it must have less rigidity than the bascule of two parts only. When one leaf only was loaded with heavy teams or cars very heavy latches would be required to give uniform deflection. They could hardly be relied on in a railroad bridge. The bridges at Chicago are claimed to be giving good service under street traffic.

The Folding Swing.—This type is only applicable to short railroad draws. It has been used very successfully in the Boston yard of the B. & M. R. R. As there is no opportunity to use lateral bracing; the trusses must be wide enough to be rigid of themselves. The limiting length of span would probably not exceed 75 ft.

The Rolling Lift.—The space required back of the abutments and the rather complicated approaches are drawbacks to the use of this form of draw. Very heavy latches must be provided at the centre or the ends on the masonry must be anchored down or counterweighted.

At Queensbury, England, there is a bridge of this type with an overhang of 60 ft. Counterweights are used. The draw rolls back underneath the fixed span, there being a movable platform which revolves a few degrees in an arc of a circle and cants the draw sufficiently to allow it to move under the floor of the shore span. Wheels 4 ft. 4 in. in diameter carry the draw as it moves back. To lift the platform hydraulic rams 8 in. in diameter are used. The rams carry a pressure of 750 lbs. per square inch.

SWING DRAW-SPANS.

Types.—When a span of greater length than about 150 ft. is required some type of riveted truss or pin-connected span is usually employed. For spans between 150 and 200 ft. the riveted truss is steadily growing in favor and in cases where the dead load is light in proportion to the live load it has decided advantages over the pin-connected truss in the direction of greater rigidity, and it may also possess a lesser advantage in economy of shop construction in certain cases. On the other hand, the field expense as well as the time required for the erection of the riveted truss will be much greater. When the dead load is large in proportion to the live load there is usually ample rigidity in either type, and the advantage of the riveted over the pin-connection exists in theory only. When a field connection is made by a great number of rivets, driven often under the most unfavorable conditions, it is a question as to what its actual efficiency may be. And a detail that requires an excess of 75 to 100 per cent to insure the original 100 per cent is surely not an ideal one. Still, as in the draw-span so many of the members require to be designed to resist compression strains or to be of built-up sections, the adoption of the riveted truss seems to follow naturally.

Lateral and Sway Bracing.—The rigidity and stiffness of the structure may be much increased by the use of stiff bracing throughout for laterals and sways. The sections used must, however, be of such size and shape that they are able to resist compression strains. Light angles used in long, unsupported lengths do not make stiff bracing (*except on the drawings*). And in any case great care should be taken that all sag or play is taken up before any riveting is done; otherwise there will be fully as much vibration and deflection as with adjustable rod-bracing, and with the added disadvantage

that any tightening up is impossible or is at best an expensive operation. A few roads are not at present using single-angle bracing except in plate-girder work, where there are no long lengths, all other bracing being made of two or more angles latticed or connected in some effective manner. This is of course expensive, but it is the only way to obtain anything that is better than rod-bracing.

Pier-panel.—There are several arrangements which may be made with that portion of the bridge which is over the centre pier. The two arms of the draw may come together in a single joint, or they may be separated by a short panel, depending in length upon the diameter of the masonry pier and the width centre to centre of trusses. In the larger draws this centre panel is usually employed, as it affords opportunity for the better distribution of the load over the masonry.

Pier Supports.—The loads may be carried down from the trusses to the pier by the centre pivot, by means of wedges driven under the trusses, by a circular row of bearing-wheels placed near the outer circumference of the pier, or by a combination of two of the above plans. Still another method which is sometimes used is to lift the span from fixed supports by means of hydraulic jacks, when the draw is to be opened, to such height as may be necessary to allow it to swing clear at the ends. This plan necessitates the lifting of the entire dead weight of the span.

Centre-bearing Draws.—In short, light spans the entire dead load, and such portions of the live load as are supported by the centre pier, are often carried entirely by the centre pivot, disks, or nest of rollers, as the case may be, the span being prevented from tilting as it is turned by four or more balance-wheels which roll on a track near the outer edge of the pier. These wheels are not assumed to carry any load except such as might be thrown upon them by an unbalanced condition of the dead load, or from wind, etc. A draw with

such an arrangement of the centre is called a *centre-bearing draw.*

Wedge-bearing Draw.—In other cases the dead load is carried by the centre pivot, and a portion or all of the live load is carried by wedges or cams driven under the chords of the trusses and so adjusted as to receive any desired amount of the load. This arrangement will be designated as the *wedge-bearing draw.*

Rim-bearing Draw.—A third plan is to carry a portion of the load by the centre pivot and the remainder by the circular row of wheels which are arranged to run between a track bolted to the pier and a circular girder (called the 'drum') which supports the span. This is called a *rim-bearing draw.*

Two Spans Continuous.—When there is no centre panel over the pier the bridge when closed is a continuous girder of two spans, and the shears, reactions, and moments are determined by the same methods as used for plate-girder draws.

Three Spans.—When the arms are separated by the panel over the pier the bridge becomes a three-span continuous girder. There are three cases to be considered: *First.* When the centre panel has no diagonal bracing. In this case, as there are no web-members in the centre panel, there can be no shear transferred across the pier (there being no members running in a diagonal direction to carry it). Thus in Fig. 1

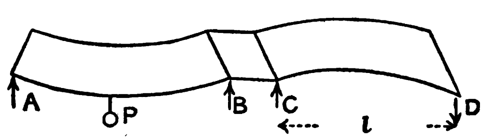


FIG. 1.

if there is a load at P in the left arm the truss will tend to take the shape shown, the rectangle at the centre becoming a parallelogram. In accordance with what we have just stated, the reactions at A and B together must be equal to P.

There will be a tendency for the end at D to rise off its support and the force necessary to hold it down will be equal to the reaction at C. As there is no load on the second arm and the sum of the vertical forces must be equal to O, then C and D must equal each other and must be of opposite sign. D is equal to the centre moment produced by the load P divided by (l) the length of one arm. If the reactions at A, B, and D be found from Table E, page 111, the strains in the various parts of the truss can be found as in the case of a two-span draw (see page 125).

Second Case. The second case to be considered is where heavy bracing is used in the centre panel. A load at any point in either arm, as at A in the left half, tends to cause the bridge to assume the position shown in Fig. 2. The arrange-

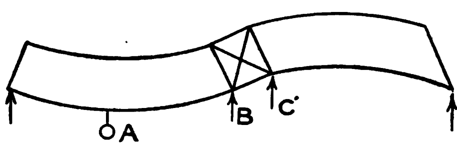


FIG. 2.

ment of the loads may be such as to lift the point C entirely off its support. This is of course objectionable, owing to the heavy blow given the structure, as it drops back to a bearing each time after the passage of the load. Where rollers or wheels are used there is no feasible method of anchoring down the points B and C, which is the only way of overcoming the trouble. There is also more or less uncertainty as to how the strains are distributed through the various members under this arrangement of the bracing. In view of the above points it is advisable to omit the bracing in the centre panel, except in cases where the reactions from uniform dead load are so great that there is no danger of there being an equal or greater negative reaction from a partial live load.* The reactions produced at A and D by a load at any point on either arm

* It is desirable however, to use light rods for purpose of adjustment.

will be positive, or upward, and the two reactions at the centre pier will be, one positive and the other negative, the positive one always being the one nearest the load. In heavy city bridges with paved floor, where the dead load is large in proportion to the live load, there would probably be economy in the use of this type and little danger of negative reactions.

Third Case. The third case which might be found advantageous where the pier supports are formed by means of cams or wedges, is shown in Fig. 3. This arrangement is called

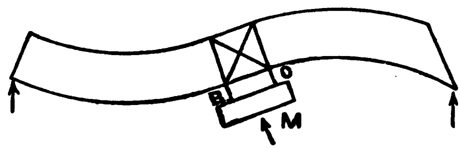


FIG. 3.

the tipper. The bridge is first carried by two points of support under each truss (*B* and *C* in the figure). These points rest upon a cantilever, which is in turn held up by the wedges or cam-bearings. In this plan the reaction at *B* and *C* must be equal, otherwise the cantilever (*m*) is not in equilibrium. To make these reactions equal there must be bracing in the centre panel, sufficiently heavy to transfer a shear equal to *B* or *C*, under a partial load.

Which type of bridge would be used in any particular case would depend upon the local conditions and to some extent upon the individual preference of the designer.

Rim and Centre Bearings.—The rim-bearing centre distributes the load over the pier better than either the centre- or wedge-bearing types, but is much more expensive. It also requires greater depth from base of rail to top of masonry, and in certain cases this is an important item. The centre-bearing span will turn much easier than a rim-bearing span of the same weight. The arrangement of centre bearing for dead load and wedge-bearing for live load is perhaps the most satisfactory plan for spans under 300 ft. in length.

Length of Centre Panel.—In a single-track through span the length of the centre panel is usually from 15½ to 16 ft., its length being made the same as the width centre to centre of trusses, the arrangement of the supporting girders, drum, etc., being somewhat simpler when the four points of truss support are at the corners of a square.

Preferable Form of Truss.—The form of truss that is generally accepted as the best at the present time is the triangular, with vertical posts and either parallel or inclined chords, there being only one set of diagonals in any panel except the centre one over the pier, none of the diagonals

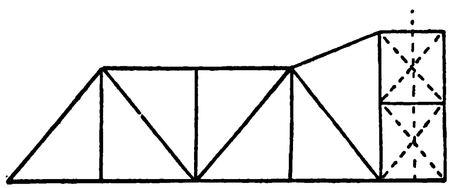


FIG. 4.

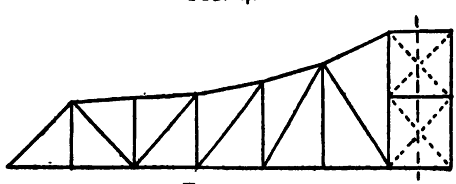


FIG. 4*a*.

in other panels being adjustable. Figs. 4 and 4*a* show examples.

Where there are counter-stresses the members are designed to resist both tension and compression. The curved top chord gives a more graceful appearance for long spans and the greater depth allowed is more economical.

Depth of Truss.—In short through spans the necessary head-room required will determine the height of truss to be used—say 26 to 30 ft. for railroad spans and 21 to 25 ft. for highway spans. Deck-spans or long through spans usually have a depth at end of $\frac{1}{8}$ to $\frac{1}{8}$ the length of one arm, and at the centre a depth of $\frac{1}{8}$ to $\frac{1}{4}$ the length of one arm.

Arrangement of Floor-beams and Stringers. — The floor-beams should rivet to the posts above or below the chords, and the stringers should frame between and be riveted to the webs of the floor-beams. Angle-bracing should be used between the stringers in the plane of the top flanges, and where possible the lateral bracing should connect to the stringers at the points where they intersect.

Centre Distributing Girders.—The distributing girders should be so designed as to deliver the weight at as many points on the drum as possible. By the arrangement shown in Plate E there are eight bearing-points on the drum, and in spans of any considerable weight there should not be less than this number. Draw-spans are often made with but four bearing-points on the drum, but this cannot be considered good practice, and the conditions seldom are such that eight points cannot be provided.

The Drum.—The main features to receive attention in the drum are to make it of such depth and section that it will distribute its loads equally over the rollers and to provide interior bracing sufficient to hold it to a true circle. If the drum be figured as a beam of length equal to the distance between the points of loading, measured on the curve, and is considered as uniformly loaded, with ends supported, but *not fixed*, it will be sufficiently heavy. Its depth ought to be about one third or more of the distance between points at which its flanges are supported by radial girders. For light spans a twenty-four (24) inch rolled beam makes a very satisfactory drum.

Treads.—The upper tread should, when possible, be riveted to the drum and then the whole surface for the roller-bearing turned up at one setting. There are several mills in the country where circles 25 to 40 ft. radius can be finished in this manner. As good results can be obtained in no other way. When this cannot be done the tread-segments should be finished separately, the drum assembled in an

inverted position, levelled up carefully, and the segments of tread then put on, levelled, and a space of about one quarter inch which has been left between them and the drum run in with melted lead or Babbitt metal; with care a good surface can be secured in this way.

The best bottom tread is unquestionably the one made and patented by the Detroit Bridge Co. It is composed of a number of flat bars set on edge and so arranged that only one of them splices at the same point. As perfect a bearing as it is possible to secure is obtained by this method, especially as the whole must be surfaced at once. This tread is shown in Fig. 35.

Bearing-wheels or -rollers.—The most common defect in the design of turntable-wheels has been due to mistaken economy. There has been too little metal used. The nearer these wheels approach to solid masses the better; any attempt at saving metal by coring out of webs should be avoided. Several types of wheel are shown and they will be more fully discussed under Machinery. The wheel shown in Plate J was used on an important draw, which was so situated that the wheels were frequently under water, especially during heavy storms in winter, and the interior of the wheel was constantly about one third full of ice or water. The wheel shown in Fig. 35 should be used on all but the lightest work, care being taken to have the rim well supported at the edges. It is not possible to keep the tracks perfectly true and the load will at times be thrown on the edges, and there will be more or less trouble with broken flanges if these are thin or not well supported.

Some years ago an arrangement for setting the wheels with the upper face or line of bearing horizontal and the axis of the wheels inclined was patented by Theodore Cooper. By this plan the shop-work is simplified by having a plane surface to finish on the drum-tread, and the great advantage is gained that if any unequal load forces the drum out of a true circle

the wheels simply slide in or out a trifle and the full bearing on all the wheels is still maintained. For illustration of this mehod see Fig. 20*a*.

Weight of Draw-spans.—The following table gives the approximate weight of railroad spans of lengths between 75 and 400 ft. Any rule or table can of course be only a mere approximation for use in preliminary design. The weights of a number of bridges as actually constructed are also given.

An approximate rule for the weight of draw-spans is: For single-track railroad draw-spans,

Deck plate girders .... $W = 9l + 120$
Lattice girders...................... $W = 7l + 200$
Pin-connected spans .... $W = 6l + 350$

l = length in feet; W = weight per foot.

These weights do not include wood floor. For double-track spans increase these weights 85 per cent.

TABLE A.

TABLE OF APPROXIMATE WEIGHT OF DRAW-SPANS, NOT INCLUDING WOOD FLOOR.

Kind.	Length.	Weight per Lin. Ft. of Bridge Proper.	Increase for Each 10 Ft. Increase of Length.	Weight of Drum, Centre Girders, and Machinery Supports.	Weight of Centre Supporting Girders, Machinery Supports, etc., where Drum is Not Used.	Weight of Machinery where Rim-bearing Centre is Used.	Weight of Machinery with Wedge or Centre Bearing.
		W		Wd		Wm	Wm
Deck pl. girder........	75	450	60	....	5000–9000	....	18,000–20,000
" " "	150	300	....	....	" "	....	" "
Thro. pl. girder........	75	675	50	....	6000–10,000	....	20,000–22,000
" " "	150	1125	....	....	" "	....	22,000–25,000
Single-track truss-span.	150	800	40	25,000–45,000	15,000–18,000	40,000–43.000	" "
" " " "	250	1200	30	30,000–50,000	18,000–25,000	43,000–46,000	23,000–26,000
" " " "	400	1600	....	40,000–70,000	25,000–40,000	45,000–47,000	24.000–28,000
Double-track truss-span	150	1500	70	35,000–55,000	....	60,000–65,000	30,000–40,000
" " " "	250	2200	60	50,000–65,000	....	65,000–73,000	....
" " " "	400	3000	....	65,000–80,000	....	70,000–80,000	....

NOTE.—The above table is for loading of 100-ton engines and train of 3000 pounds per lineal foot.

For different loads use proportional weights.

TABLE B.

WEIGHT, COST, AND MOTIVE POWER OF EXISTING DRAW-SPANS.

Kind.	Length.	Width.	Capacity.	Weight.	Cost.	Power.	
	Ft.	Ft.		Lbs.			
Highway	70	15	100 pounds per square foot	35,000	$1,500	Hand	Lattice girder
S. T. R. R.	82	7	112-ton engine, 3000 pounds	48,000	4,000	"	Deck pl. girder
City, 20′ rdy., two 8′ walks	97	36				One 10 H. P. M.	
" 41′ " " 7′ "	88	55	100 pounds + dead		42,000	Two 25 H. P. M.	Double lift
" 42′ " " 8′ "	115	58			169,000	" 50 "	" " rolling
" 40′ " " 10′ "	130	60	4500 live, 4000 dead, per foot	Total 1,200,000 Span 500,000	190,000	One 115 H. P. S.	Vertical "
R. R., single-track	148	16	104-ton engine, 3000 pounds	145,000		Hand	Through Pratt
" " "	143	15½	95 " " 3000 "	165,000	7,500	"	Half-through pl. girder
" " "	119	6½	4000 pounds per lineal foot	100,000	4,000	"	Deck pl. girder
Highway, 35′ rdy., two 5′ walks	146	45	100 pounds + dead	310,000	47,000	"	" " " sliding
R. R., single-track	150	15½	112-ton engine, 4000 pounds	163,000		"	Through Pratt
" double "	160	29	" " " " "	419,000	20,000	"	" "
" single "	164	16	107 " " 3000 "	110,000		"	" "
" " "	179	16	104 " " " "	159,000		"	" "
" " "	182	16	96 " " " "	176,000		"	" "
" " "	186	16	112 " " 4000 "	202,000		"	" curved top chord
Highway, 29′ rdy., two 10′ walks	180	29		276,000		"	" " " "
" 21′ " one 5′ walk	196	26		98,000		"	" " " "
City, 41′ rdy., two 10′ walks	199	61		410,000		"	" " " "
" 26′ " " 8′ "	314	54		370,000		"	" " " "
R. R., single-track	200	16	Cooper, class A	170,000		"	" " " "
" " "	200	16	" " B	150,000		"	" " " "
" " "	210	16	104-ton, 3000 pounds	204,000		"	" " " "
" " "	250	16	126 " 4000 "	328,000		"	" " " "
" " "	256	16	Cooper, class A	230,000		"	" " " "
" " "	260	16	" " "	272,000		"	" " " "
" " "	287	17	107-ton engine, 4000 pounds	357,000		"	" " " "
" " "	300	17	Cooper, class A	500,000		One 12 H. P. S.	" " " "
" " "	356	16	104-ton, 3000 pounds	683,000		" 15 "	" " " "
" " "	290	16	95 " " "	480,000	17,500	" 12 H. P. G.	" " " "
" double "	247	29	120 " 4000 "	570,000	38,000	" 15 H. P. S.	" " " "
" four "	390	63	12,000 live, 6950 dead	5,000,000	218,000	Two 50 "	" " " "
Highway and R. R., double-track	365	29	12,000 " 7400 "	2,467,000			" " " "
" " " single "	470	29		3,000,000			" " " "

Table B. (*Continued.*)

Kind.	Length.	Width.	Capacity.	Weight.	Cost.	Power.	
	Ft.	Ft.		Lbs.			
Highway	140	12	80 pounds per square foot		$6,700	Hand	Double swing
R. R., four tracks	176	57	12,000 pounds live, 7000 dead	1,100,000		One 25 H. P. M.	Through pl. girder
City	279	30	3000 pounds + dead	854.000	35,000	Two 25 "	Double swing
Highway and R. R.	520		2 R. R. tracks, 2 highways	3,000.000		" 40 "	
Highway, city	270			1,800,000		One 40 H. P. S.	
" "	400			4,200,000		" 50 "	
R. R., single-track	362			684.000		" 20 "	
" double "	217			600,000		" 20 "	
" " "	300			1,250,000		" 20 "	
" " "	400			2,000,000		" 35 "	
" " "	500			2,400,000		" 40 "	
" single "	340	16	Cooper, 1896, E	815,000	14.500	" 30 "	Through curved chord
" " "	318	16	Mobile & Ohio	973,000		Hand	" " "
DESIGNS FOR THE DRAW OVER THE DULUTH SHIP-CANAL.							
Design 1	250				236,000	116 H. P. S.	Sliding, single
" 2	250				80,000		Pontoon, single
" 3	250				81,000	8 H. P. S.	Sliding, single
" 4	250				170,000		Double swing
" 5	250				108,000		Pontoon, single
" 6	250				190,000		Double swing
" 7	250				125,000		Single "
" 8	250				125,600		Double lift
" 9	250				140,000		Sliding, single
" 10	250				150.000		" double
" 11	250				125,000		Vertical lift
" 12	250				154,000		Sliding, double
R. R., single-track	150	7	Two 125-ton engine + 3500	160.000	6,000	Hand	Deck pl. girder
City, 60′ rdy., two 12′ walks	321	60		2,400,000			Curved lattice
" 40′ " " " "	200		100 pounds per square foot	418,000			" through

ESTIMATED COST.

Double-track span	369′ 0″	Total cost superstructure	$45,000	Machinery	6 cents per pound
" " "	480′ 0″	" " "	51,000	"	5½ " " "
Eight-track swing	400′ 0″	" " "	315,000	"	$80,000
" " rolling	400′ 0″	" " "	260,000	"	76,000
" " lift	400′ 0″	" " "	347,000	"	93,000

General Formulæ.—The general formulæ necessary for the computation of the stresses in the trusses are given below:

1st. Two-span Draws.—a is the distance of the load P from left support; $K = \frac{a}{l}$; $l =$ length of one arm.

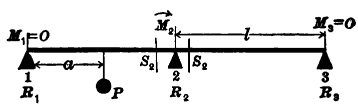


FIG. 5.

Let R_1, R_2, R_3 represent the reactions at the three points of support; S_1, S_2', S_2, S_3 represent the shears in the truss just to the right or left of the supports; and let M_1, M_2, M_3 represent the moments at the supports. Then for a load P at any point in the left arm we have

$$R_1 = S_1 = \frac{P}{4}(4 + K^3 - 5K); \quad . \quad . \quad . \quad . \quad (1)$$

$$R_2 = S_2' + S_2 = \frac{P}{2}(3K - K^3); \quad . \quad . \quad . \quad (2)$$

$$R_3 = S_3 = \frac{P}{4}(K^3 - K); \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$M_1 = 0; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$M_2 = -\frac{Pl}{4}(K - K^3); \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$M_3 = 0. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

For loads in the right arm the formulæ are of course the same except reversed in order.

2d. Three Spans Continuous.

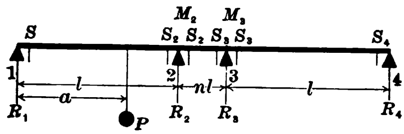


FIG. 6.

$$R_1 = S_1 = \frac{P}{H}[H - HK - (K - K^3)(2 + 2n)]; \quad . \quad . \quad (1)$$

$$R_2 = S_2' + S_2 = \frac{P}{H}\left[HK + (K - K^3)\left(5 + 2n + \frac{2}{n}\right)\right]; \quad (2)$$

$$R_3 = S_3' + S_3 = -\frac{P}{H}\left[(K - K^3)\left(3 + n + \frac{2}{n}\right)\right]; \quad . \quad (3)$$

$$S_4 = \frac{P}{H}(K - K^3)n = R_4; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$S_2' = \frac{P}{H}[HK + (K - K^3)(2 + 2n); \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$S_2 = \frac{P}{H}\left[(K - K^3)\left(3 + \frac{2}{n}\right)\right]; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$S_3' = -S_2; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$S_3 = -S_4. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$H = 4 + 8n + 3n^2$; $K = \frac{a}{l}$; and n = ratio of length of centre span to length of end spans, as $\frac{1}{6}$, $\frac{1}{8}$, etc.

The formulæ for the moments are $M_1 = M_4 = 0$.

$$M_2 = -\frac{2P(K - K^3)(l + nl)}{H}; \quad . \quad . \quad . \quad (9)$$

$$M_3 = \frac{P(K - K^3)nl}{H}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

Equations (1), (4), and (6) are all that are required when the two end spans are of equal length.

3d. Three Spans Partially Continuous.—In this case the centre span is without diagonal bracing, or if bracing is used it is made very light and is not considered in determining the stresses. There being no web-bracing, there can be no shearing-forces carried across the pier or through the centre panel. The result of this is that the moments at 2 and 3 must be equal, and the formulæ for this case are:

$$R_1 = S_1 = P\left[1 - K - \frac{1}{2+3n}(K - K^3)\right]; \quad . \quad . \quad . \quad (1)$$

$$R_2 = S'_2 = P - R_1; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$R_3 = S_3 = -S_4 = -R_4 = -P\left[\frac{1}{2+3n}(K - K^3)\right]; \quad (3)$$

$$M_2 = M_3 = Pl\frac{(K - K^3)}{2+3n} = R_1 l - Pl(1 - K). \quad . \quad . \quad (4)$$

This case is often considered as a two-span bridge (the centre panel being assumed to be 0) and only balanced loads are used. For all unbalanced loads or for unbalanced portions of loads the bridge is regarded as two independent or separate spans. When the ends of the span are raised this assump-

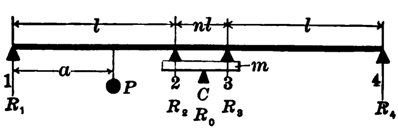


FIG. 7.

tion is not strictly correct, however. Where the centre panel is long in proportion to the length of the arms a safer method would be to find the stresses on both assumptions and use the greater.

4th. The Tipper.—If the cantilever under the centre panel be considered as a part of the truss this case is the same as a

two-span bridge and the formulæ as given for that condition may be used. The reactions at 2 and 3 must be equal, otherwise the cantilever m is not in equilibrium. Bracing must then be used in the centre panel sufficient to transfer a shear equal to R_2 or R_3, which is one half of R_c.

If the assumption is made that the bridge is a three-span continuous girder with equal reactions at 2 and 3 in all cases the formulæ become:

$$R_1 = S_1 = \frac{P}{2H}[2H - (10 + 15n + 3n^2)K + (2 + n)K^2]; \quad (1)$$

$$R_2 = R_3 = S_2' + S_2 = S_3' + S_3$$
$$= \frac{P}{2H}[(6 + 9n + 3n^2)K - (2 + n)K^2]; \quad (2)$$

$$S_2' = \frac{P}{2H}[(10 + 15n + 3n^2)K - (2 + n)K^2)]; \quad . \quad . \quad . \quad . \quad (3)$$

$$S_2 = -\frac{P}{2H}(4 + 6n)K, \quad S_3' = -S_2; \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$S_3 = -S_4 = \frac{P}{2H}[(2 + 3n + 3n^2)K - (2 + n)K^2]; \quad . \quad . \quad (5)$$

$$R_4 = S_4 = -S_3. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The formulæ given above are based upon the theory of constant moment of inertia. Investigations have recently been made to see what changes would be produced by the use of a variable moment of inertia. See "The Continuous Girder," by M. A. Howe.

It is found that the results obtained are not essentially different and the formulæ are much more cumbersome and tedious to use.

The formulæ as derived for the two-span structure are given below:

$$M_2 = \frac{\Sigma_a^l (x-a)\dfrac{\Delta x}{I_x} - \Sigma_0^l x \dfrac{\Delta x}{I_x}(l-a)P_2}{3l\Sigma_0^l \dfrac{\Delta x}{I_x} - 2\Sigma_0^l x \dfrac{\Delta x}{I_x}}.$$

I_x is the moment of inertia at various points;
Δx to be assumed some small fraction of the length, say $\frac{1}{20}$;
Σ signifying the summation of the various terms between the limits given;
M_2 being found, the reactions are readily determined from the usual formulæ.

Twin or Double Drawbridges.—The arrangement shown in Fig. 8 can be made partially continuous by suitable backing

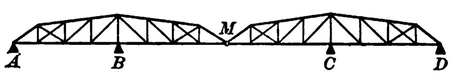


FIG. 8.

arrangement at M. There can be no moment transferred by the joint, but shearing-forces may be carried across.

The various reactions for a load on the arm AB are:

$$A = \frac{P}{8}(8 - 9K + K^3);$$

$$B = \frac{P}{8}(10K - 2K^3);$$

$$C = -\frac{P}{8}(2K - 2K^3);$$

$$D = \frac{P}{8}(K - K^3).$$

For a load on arm BM the reactions are:

$$A = -\frac{P}{8}(6K - 3K^2 + K^3);$$

$$B = \frac{P}{8}(8 + 4K - 6K^2 + 2K^3);$$

$$C = \frac{P}{8}(4K + 6K^2 - 2K^3);$$

$$D = -\frac{P}{8}(2K + 3K^2 - K^3).$$

For the arrangement shown in Fig. 9 the reactions are: For load on arm AB,

$$A = \frac{P}{8 + 12n}(8 + 12n) - (9 + 12n)(K + K^3);$$

$$B = P - A;$$

$$C = F = PK - A;$$

$$D = E = -PK + A.$$

For load on arm CM,

$$-A = B = \frac{P}{8 + 12n}(6 + 6n)(K - 3K^2 + K^3);$$

$$C = B + P(1 - K);$$

$$D = E = -F = PK - B.$$

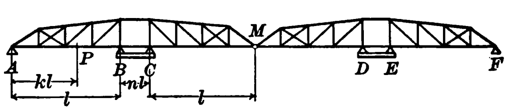


FIG. 9.

Unequal Arms. — When the lengths of the arms are unequal the shorter one must be counterweighted until they

balance. The more nicely this is done the easier the span will turn. The use of the formulæ for three continuous spans of unequal length will be necessary in determining the stresses unless such provision be made for lifting the ends as will insure the two arms acting as independent spans when the draw is closed. Wherever possible this plan should be adopted, as it will give a more satisfactory structure in every respect. Several plans for accomplishing this will be described later.

Tables and Diagrams.—Much labor may be saved by tabulating the coefficients in the formulæ for shears and moments, as has been done in the tables which follow. These tables have been made for single- and double-track spans, between such limits as will include the majority of cases likely to occur.

Coefficients for Shears and Moments.—If we let l represent the length of one arm of the draw and l_1 the length of the panel over the pier the ratio will be $l_1 \div l$. For a span 166 ft. long $l = 75$ and $l_1 = 16$; then $\frac{l_1}{l} = 0.21$. For a span of 416 ft. $l = 200$, $l_1 = 16$, $\frac{l_1}{l} = 0.08$. An average span will be, say, 256 ft., where $l = 120$ and $l_1 = 16$; $\frac{l_1}{l} = .133$. As the values of the coefficients used in the determination of shears and moments depend upon the ratio of $\frac{l_1}{l}$, tables have been prepared which give the values of the coefficients for three values of $\frac{l_1}{l}$. A comparison of these tables (see page 108) will show that the average values for $\frac{l_1}{l} = .133$ do not vary more than .007 from either of the others. This table of average values may then safely be used for all spans between

150 and 450 ft. in length when the centre panel does not exceed about 16 ft.

In double-track spans where the length of centre panel is 28 to 30 ft. the values of the coefficients will be so different that separate tables have been prepared. The reactions are all that are necessary to determine the stresses in the various parts. The simplest method is perhaps the graphical one. This and several others are given in any text-book on stresses and space will not be taken here to explain them.*

Tables of strains for a 293-ft. span are given on page 126, and diagrams will be found at the back part of the book.

TABLE C.

COEFFICIENTS D_1 AND C_2 FOR THREE-SPAN BRIDGE. CENTRE PANEL BRACED.

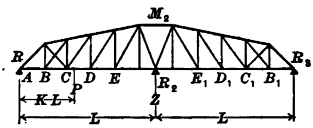
K	$K - K^3$	C_2 for $n = .1666$	C_2 for $n = .10$	C_2 for $n = .1333$	$1 - K$	D_1 for $n = .1666$	D_1 for $n = .10$	D_1 for $n = .1333$
0.0	0.000	0.000	0.000	0.000	1.0	1.000	1.000	1.000
0.1	0.099	0.039	0.043	0.041	0.9	0.861	0.857	0.859
0.2	0.192	0.077	0.084	0.080	0.8	0.723	0.716	0.720
0.3	0.273	0.109	0.119	0.114	0.7	0.591	0.581	0.586
0.4	0.336	0.134	0.146	0.140	0.6	0.466	0.454	0.460
0.5	0.375	0.150	0.163	0.156	0.5	0.350	0.337	0.344
0.6	0.384	0.154	0.167	0.160	0.4	0.246	0.233	0.240
0.7	0.357	0.143	0.155	0.149	0.3	0.157	0.145	0.151
0.8	0.288	0.115	0.125	0.120	0.2	0.085	0.075	0.080
0.9	0.171	0.068	0.074	0.071	0.1	0.032	0.026	0.029
1.0	0.000	0.000	0.000	0.000	0.0	0.000	0.000	0.000

K = distance from right-hand or left-hand support to a load P.
C_2 = coefficient for the moment at centre pier.
D_1 = coefficient for the reaction at end of loaded arm.
$n = \frac{l_1}{l}$. l = length of one arm. l_1 = length of centre panel.

* See Merriman, Burr, DuBois, or Johnson.

TABLE D.

TWO SPANS CONTINUOUS.



LOAD IN FIRST ARM.	LOAD IN SECOND ARM.
$M_2 = C_1P_1L$;	$M_2 = C_2P_2L$;
$S_1 = D_1P_1$;	$S_1 = D_2P_2 = C_2P_2$;
$X_0 = E_1L$;	$X_0 = 0$.

EXPLANATION.

M_2 = moment at centre pier.
C_1 = coefficient taken from diagram for loads in first arm.
C_2 = coefficient taken from diagram for loads in second arm.
P_1 = load at any point in first arm.
P_2 = load at any point in second arm.
L = length of arm on half-span.
S_1 = shear at abutment.
D_1 = coefficient given by diagram for load in first arm.
D_2 = coefficient given by diagram for load in second arm.
E_1 = point of o moment in first arm.
X_0 = distance from left abutment to point of o moment.

COEFFICIENTS C_1 FOR LOADS IN FIRST ARM AND COEFFICIENTS C_2 AND D_2 FOR LOADS IN SECOND ARM.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.0586	.0938	.0820						.2344
5	.048	.084	.096	.072					.300
6	.0406	.0740	.0937	.0925	.0637				.3645
7	.0350	.0656	.0875	.0962	.0875	.0568			.4285
8	.0308	.0586	.0806	.0938	.0952	.0820	.0513		.4923
9	.0274	.0527	.0740	.0891	.0960	.0925	.0767	.0466	.5550

COEFFICIENTS D_1 FOR LOADS IN FIRST ARM.

4	.691	.406	.168						1.265
5	.752	.516	.304	.128					1.700
6	.792	.592	.406	.241	.103				2.134
7	.822	.649	.484	.332	.198	.086			2.571
8	.844	.691	.544	.406	.280	.168	.074		3.007
9	.861	.725	.592	.466	.348	.241	.146	.0465	3.444

VALUES OF E_1 FOR LOADS IN FIRST ARM.

4	.810	.842	.900						
5	.807	.827	.862	.916					
6	.805	.818	.842	.879	.929				
7	.803	.813	.830	.856	.893	.943			
8	.803	.810	.824	.842	.868	.900	.943		
9	.801	.809	.820	.832	.852	.879	.910	.950	

TABLE D. (*Continued.*)

LOADS FOR MAXIMUM NEGATIVE MOMENTS—FIRST ARM.

4	—	—	—	—	—	—	— — —
5	—	—	—	—	—	—	— — —
6	*B*	*C*					For maximum at *F*
7	*B*	*C*	*D*	*E*			" " " *G*
8	*B*	*C*	*D*	*E*	*F*		" " " *H*
9	*B*	*C*	*D*	*E*	*F*	*G*	" " " *I*

All loads on second arm in each case. All loads cause negative moments over pier.

LOADS FOR MAXIMUM POSITIVE MOMENTS—FIRST ARM.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
									Max. at
4	*B*	*C*	*D*						*B* to *D*
5	*B*	*C*	*D*	*E*					*B* to *E*
6	*B*	*C*	*D*	*E*	*F*				*B* to *E*
6			*D*	*E*	*F*				*F*
7	*B*	*C*	*D*	*E*	*F*	*G*			*B* to *F*
7					*F*	*G*			*G*
8	*B*	*C*	*D*	*E*	*F*	*G*	*H*		*B* to *G*
8						*G*	*H*		*H*
9	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*B* to *H*
9							*H*	*I*	*I*

SHEARS: All loads on second arm cause negative shear in first arm.
Loads moving *A* towards Z cause negative shear in first arm.
Loads moving *Z* towards *A* cause positive shear in first arm.

P_1 = any load in first arm.
P_2 = any load in second arm.
S_1 = reaction at *A* from P_1 or P_2.
M_2 = moment at pier from P_1 or P_2.
X_0 = distance from *A* to point of zero moment in first arm.
L = length of half-span.

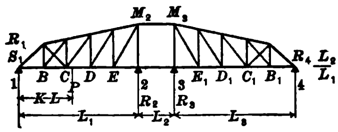
$M_2 = C_1P_1L$ or C_2P_2L.
$S_1 = D_1P_1$ or D_2P_2.
$X_0 = E_1L_1$.

WEB-STRESSES: Max. stress in any $\left\{\begin{matrix}\text{mem.,}\\\text{web,}\end{matrix}\right\}$ load moving *A* to *Z*, is when load extends from *A* to piece in question, and full load on 2d arm.

Max. stress in any $\left\{\begin{matrix}\text{mem.,}\\\text{web,}\end{matrix}\right\}$ load moving *Z* to *A*, is when load extends from *Z* to piece in question, and no load on 2d arm.

TABLE E.

FOUR SUPPORTS. NO BRACES IN CENTRE PANEL.



$L_1 = L_3$; $\qquad \frac{L_2}{L_1} = n = 0.1333$;

$M_2 = M_3 = -C_2PL$; $\qquad C_2 = \frac{K - K^3}{4 + 6n}$;

$R_3 = R_4 = C_2P$; $\qquad .208(K - K^3)$;

$S_1 = D_1P$;

$R_2 = P - S_1$; $\qquad D_1 = 1 - K - \frac{K - K^3}{4 + 6n} = 1 - K - .208(K - K^3)$.

$X_0 = EL =$ distance to point of zero moment.

$$E_1 = \frac{K}{1 - D_1}.$$

COEFFICIENTS – C_2.

Number of Panel.	B	C	D	E	F	G	H	I	Total.
4	.0490	.0780	.0686						.1956
5	.0401	.0700	.0800	.0600					.2501
6	.0340	.0618	.0780	.0773	.0530				.3041
7	.0294	.0550	.0730	.0801	.0730	.0480			.3585
8	.0260	.0490	.0671	.0780	.0793	.0686	.0435		.4115
9	.0231	.0442	.0618	.0742	.0800	.0773	.0640	.0395	.4641

COEFFICIENTS – D_1.

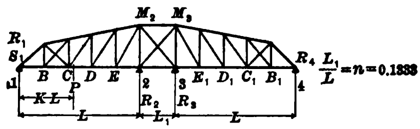
Number of Panel.	B	C	D	E	F	G	H	I	Total.
4	.698	.420	.180						1.298
5	.758	.525	.320	.140					1.743
6	.796	.600	.420	.254	.113				2.183
7	.825	.656	.493	.348	.210	.094			2.626
8	.845	.698	.553	.420	.292	.180	.082		3.070
9	.862	.732	.600	.475	.361	.254	.156	.072	3.512

COEFFICIENTS – E_1.

Number of Panel.	B	C	D	E	F	G	H	I	Total.
4	.834	.866	.914						
5	.831	.851	.882	.930					
6	.830	.842	.866	.894	.940				
7	.829	.837	.854	.877	.905	.946			
8	.828	.834	.848	.866	.886	.914	.953		
9	.828	.832	.842	.858	.874	.894	.922	.958	

TABLE F.

THREE SPANS CONTINUOUS.



$$M_2 = - C_2 PL; \qquad C_2 = 0.443(K - K^3);$$
$$M_3 = C_3 PL; \qquad C_3 = 0.026(K - K^3);$$
$$S_1 = D_1 P; \qquad D_1 = (I - K - 0.443(K - K^3));$$
$$S_2 = D_2 P; \qquad D_2 = 3.515(K - K^3);$$
$$S_4 = C_3 P; \qquad R_4 = S_4;$$
$$R_2 = F_2 P; \qquad R_1 = S_1;$$
$$R_3 = - F_3 P; \qquad F_2 = K + 3.958(K - K^3);$$
$$F_3 = 3.542(K - K^3).$$

COEFFICIENTS – D_1.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.645	.337	.105						1.087
5	.715	.451	.232	.071					1.469
6	.760	.538	.338	.170	.053				1.859
7	.794	.600	.418	.260	.132	.040			2.244
8	.820	.645	.480	.338	.205	.105	.034		2.627
9	.836	.683	.538	.396	.275	.171	.088	.028	3.015

COEFFICIENTS – D_2.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.820	1.318	1.150						3.288
5	.670	1.183	1.350	1.010					4.213
6	.560	1.040	1.318	1.300	.886				5.104
7	.490	.920	1.230	1.352	1.230	.795			6.017
8	.431	.820	1.133	1.318	1.340	1.150	.720		6.912
9	.390	.740	1.040	1.255	1.351	1.300	1.080	.650	7.806

COEFFICIENTS – F_2.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	1.18	1.98	2.05						5.21
5	.96	1.73	2.12	1.94					6.75
6	.80	1.52	1.98	2.13	1.83				8.26
7	.69	1.32	1.82	2.10	2.10	1.74			9.77
8	.62	1.18	1.65	1.98	2.13	2.05	1.68		11.29
9	.56	1.06	1.51	1.86	2.08	2.13	2.00	1.62	12.82

COEFFICIENTS – F_3.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.835	1.330	1.162						3.327
5	.685	1.194	1.360	1.020					4.259
6	.572	1.054	1.330	1.310	.890				5.156
7	.500	.930	1.240	1.363	1.240	.800			6.073
8	.440	.835	1.145	1.330	1.348	1.162	.725		6.985
9	.400	.750	1.054	1.268	1.362	1.310	1.090	.660	7.894

TABLE F. (*Continued.*)

COEFFICIENTS C_2.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.104	.165	.146						.415
5	.084	.148	.169	.127					.528
6	.070	.131	.165	.163	.112				.641
7	.061	.116	.154	.170	.155	.100			.756
8	.054	.104	.142	.165	.168	.146	.090		.869
9	.048	.092	.131	.157	.169	.163	.136	.082	.978

COEFFICIENTS – C_3.

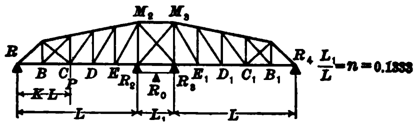
Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.0060	.0097	.0085						.0242
5	.0050	.0087	.0100	.0075					.0312
6	.0040	.0078	.0097	.0097	.0068				.0380
7	.0035	.0068	.0090	.0100	.0090	.0060			.0443
8	.0030	.0060	.0084	.0097	.0100	.0085	.0055		.0511
9	.0030	.0055	.0077	.0092	.0100	.0097	.0080	.0050	.0581

COEFFICIENTS – E_1.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.835	.864	.917						
5	.833	.852	.883	.930					
6	.831	.843	.864	.897	.940				
7	.830	.839	.855	.878	.910	.947			
8	.829	.835	.849	.864	.887	.917	.953		
9	.829	.833	.843	.858	.874	.897	.923	.958	

TABLE G.

TIPPER. THREE CENTRE SUPPORTS.



$M_2 = C_2PL$; $\qquad C_2 = 0.177K + 0.208K^3$;

$M_3 = C_3PL$; $\qquad C_3 = 0.24K - 0.208K^3$;

$S_1 = + D_1P$; $\qquad D_1 = I - 1.177K + 0.208K^3$;

$S_2 = D_2P$; $\qquad D_2 = 0.469K$;

$S_4 = - C_3P$; $\qquad R_1 = S_1$;

$R_2 = F_2P$; $\qquad R_4 = S_4$;

$R_3 = R_2$; $\qquad F_2 = 0.708K - 0.208K^3$.

COEFFICIENTS D_1.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.709	.439	.205						1.353
5	.768	.542	.339	.165					1.814
6	.805	.613	.438	.278	.140				2.274
7	.830	.667	.510	.365	.237	.122			2.731
8	.851	.710	.567	.437	.315	.205	.110		3.195
9	.868	.740	.614	.493	.382	.278	.183	.099	3.657

COEFFICIENTS $- C_3$.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.0567	.0940	.0920						.2427
5	.0461	.0831	.0990	.0852					.3134
6	.0386	.0725	.0940	.0982	.0795				.3828
7	.0333	.0635	.0866	.0981	.0953	.0745			.4513
8	.0295	.0568	.0792	.0940	.0990	.0920	.0705		.5210
9	.0262	.0508	.0725	.0883	.0976	.0981	.0888	.0670	.5893

COEFFICIENTS $- D_2$.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.115	.232	.350						.697
5	.091	.188	.280	.375					.934
6	.075	.154	.232	.311	.390				1.162
7	.066	.132	.200	.266	.333	.400			1.397
8	.057	.116	.175	.232	.293	.350	.410		1.633
9	.050	.101	.154	.206	.260	.312	.362	.416	1.861

COEFFICIENTS $- C_2$.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	−.0411	−.0625	−.0450						−.1486
5	−.0338	−.0578	−.0615	−.0350					−.1881
6	−.0284	−.0515	−.0625	−.0568	−.0270				−.2262
7	−.0248	−.0456	−.0596	−.0625	−.0508	−.0210			−.2643
8	−.0220	−.0412	−.0558	−.0625	−.0600	−.0450	−.0156		−.3021
9	−.0200	−.0370	−.0515	−.0605	−.0628	−.0566	−.0400	−.0100	−.3384

TABLE G. (*Continued.*)

COEFFICIENTS – F_2.

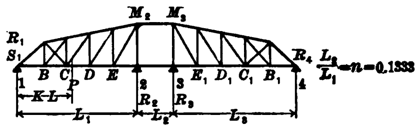
Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.172	.328	.442						.942
5	.139	.270	.379	.460					1.248
6	.115	.227	.328	.410	.470				1.550
7	.099	.195	.286	.365	.430	.475			1.850
8	.085	.172	.254	.328	.390	.442	.480		2.151
9	.078	.152	.228	.298	.355	.410	.452	.482	2.455

COEFFICIENTS E_1.

Number of Panels.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Totals.
4	.858	.887	.943						2.688
5	.855	.873	.905	.960					3.593
6	.853	.865	.887	.922	.970				4.497
7	.852	.861	.877	.900	.933	.977			5.400
8	.852	.858	.871	.887	.912	.943	.983		6.306
9	.851	.856	.865	.880	.897	.922	.952	.988	7.211

TABLE H.

DOUBLE-TRACK DRAW-SPAN. NO BRACES IN CENTRE PANEL. FOUR SUPPORTS.



$L_1 = L_3$; $\frac{L_2}{L_1} = n$; $M_2 = M_3 = -C_2PL$;

$R_1 = R_4 = C_2P$; $S_1 = D_1P$; $R_2 = P - S_1$.

$X_0 = E_1L$ = distance to point of zero moment.

VALUES OF C_2 AND D_1 FOR VARYING n.

n	C_2	D_1
0.4	$0.16\ (K - K^3)$	$1 - K - 0.16\ (K - K^3)$
0.193	$0.194\ (K - K^3)$	$1 - K - 0.194\ (K - K^3)$
0.136	$0.207\ (K - K^3)$	$1 - K - 0.207\ (K - K^3)$
0.25	$0.182\ (K - K^3)$	$1 - K - 0.182\ (K - K^3)$

$$E_1 = \frac{K}{1 - D_1}.$$

COEFFICIENTS — C_2.

No. of Panel.	B	C	D	E	F	G	H	I	Total	Values of n.
4	.0370	.0590	.0506						.1467	0.4
	.0459	.0730	.0635						.1824	.193
	.0490	.0780	.0692						.1962	.136
	.0430	.0681	.0600						.1711	.25
5	.0301	.0531	.0600	.0448					.1880	0.4
	.0373	.0650	.0748	.0560					.2331	.193
	.0400	.0700	.0800	.0614					.2514	.136
	.0351	.0613	.0700	.0530					.2194	.25
6	.0252	.0470	.0591	.0573	.0400				.2286	0.4
	.0314	.0575	.0730	.0720	.0498				.2837	.193
	.0338	.0616	.0780	.0771	.0550				.3055	.136
	.0295	.0545	.0681	.0675	.0471				.2667	.25
7	.0218	.0412	.0555	.0602	.0540	.0360			.2687	0.4
	.0274	.0511	.0680	.0750	.0675	.0440			.3330	.193
	.0298	.0548	.0730	.0802	.0730	.0488			.3596	.136
	.0257	.0481	.0640	.0701	.0639	.0414			.3132	.25
8	.0190	.0370	.0510	.0591	.0592	.0506	.0326		.3085	0.4
	.0240	.0459	.0623	.0730	.0740	.0635	.0402		.3829	.193
	.0260	.0490	.0670	.0780	.0793	.0692	.0430		.4123	.136
	.0222	.0430	.0590	.0681	.0694	.0600	.0378		.3595	.25
9	.0170	.0333	.0470	.0564	.0602	.0573	.0475	.0298	.3485	0.4
	.0215	.0414	.0575	.0692	.0749	.0720	.0595	.0372	.4332	.193
	.0232	.0443	.0616	.0740	.0800	.0771	.0655	.0400	.4657	.136
	.0200	.0389	.0545	.0650	.0700	.0675	.0565	.0340	.4064	.25

TABLE H. (*Continued.*)

COEFFICIENTS – D_1.

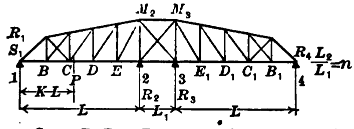
No. of Panel.	B	C	D	E	F	G	H	I	Total.	Values of n.
4	.711	.443	.200						1.354	0.4
	.703	.425	.187						1.315	.193
	.702	.422	.182						1.306	.136
	.706	.431	.190						1.327	.25
5	.770	.549	.342	.156					1.817	0.4
	.761	.535	.326	.145					1.767	.193
	.761	.531	.320	.140					1.752	.136
	.765	.539	.331	.149					1.784	.25
6	.810	.618	.443	.280	.128				2.279	0.4
	.802	.606	.425	.262	.118				2.213	.193
	.801	.602	.422	.258	.114				2.197	.136
	.805	.609	.431	.268	.122				2.235	.25
7	.832	.670	.518	.370	.231	.107			2.728	0.4
	.827	.660	.501	.353	.217	.098			2.656	.193
	.827	.658	.498	.348	.212	.093			2.636	.136
	.830	.663	.506	.358	.220	.100			2.677	.25
8	.855	.711	.574	.443	.320	.200	.093		3.196	0.4
	.851	.703	.561	.425	.301	.187	.087		3.115	.193
	.850	.702	.559	.422	.298	.182	.082		3.095	.136
	.852	.706	.564	.431	.309	.190	.089		3.141	.25
9	.871	.741	.618	.500	.388	.280	.175	.083	3.656	0.4
	.866	.734	.606	.483	.370	.262	.163	.076	3.560	.193
	.865	.732	.602	.480	.363	.258	.160	.072	3.532	.136
	.869	.735	.609	.489	.375	.268	.168	.079	3.592	.25

COEFFICIENTS – E_1.

No. of Panel.	B	C	D	E	F	G	H	I	Total.	Values of n.
4	.872	.893	.934						2.699	0.4
	.845	.873	.920						2.638	.193
	.836	.863	.913						2.612	.136
	.853	.880	.923						2.656	.25
5	.870	.882	.907	.946					3.605	0.4
	.843	.860	.888	.933					3.524	.193
	.833	.850	.881	.928					3.492	.136
	.851	.867	.894	.937					3.549	.25
6	.868	.877	.893	.917	.953				4.508	0.4
	.841	.852	.873	.900	.943				4.409	.193
	.831	.843	.863	.893	.937				4.367	.136
	.850	.860	.880	.905	.946				4.441	.25
7	.868	.873	.884	.902	.927	.960			5.414	0.4
	.840	.849	.863	.883	.911	.950			5.296	.193
	.830	.839	.853	.875	.903	.947			5.247	.136
	.850	.857	.870	.890	.915	.953			5.335	.25
8	.867	.872	.880	.893	.911	.934	.964		6.321	0.4
	.840	.845	.857	.873	.893	.920	.955		6.183	.193
	.830	.836	.847	.863	.885	.913	.952		6.126	.136
	.848	.853	.864	.880	.898	.923	.958		6.224	.25
9	.867	.870	.877	.885	.900	.917	.940	.967	7.223	0.4
	.839	.843	.852	.864	.881	.900	.927	.960	7.066	.193
	.829	.834	.843	.855	.873	.893	.921	.957	7.005	.136
	.848	.852	.860	.873	.887	.905	.941	963	7.129	.25

TABLE I.

DOUBLE-TRACK DRAW-SPAN. BRACES IN CENTRE PANEL. FOUR SUPPORTS.



$M_2 = -C_2PL$; $S_1 = D_1P = R_1$; $S_4 = C_3P = R_4$; $R_3 = F_3P$.

$M_3 = C_3PL$; $S_2 = D_2P$; $R_2 = F_2P$; $L_2 + L_1 = n$

$X_0 = E_1L$ = distance from short end to point of zero moment.

VALUES C_2. C_3, ETC., FOR VARYING n.

n	.4	.193	.136	.25
C_2	$.3646(K-K^3)$	$.4217(K-K^3)$	$.4416(K-K^3)$	$.404(K-K^3)$
C_3	$.052(K-K^3)$	$.0342(K-K^3)$	$.0265(K-K^3)$	$.040(K-K^3)$
D_1	$1-K-.3646(K-K^3)$	$1-K-4217(K-K^3)$	$1-K.4416(K-K^3)$	$1K-.404(K-K^3)$
D_2	$1.0417(K-K^3)$	$2.3579(K-K^3)$	$3.4325(K-K^3)$	$1.1778(K-K^3)$
F_2	$K+1.4063(K-K^3)$	$K+2.7795(K-K^3)$	$K+3.8741(K-K^3)$	$K+2.182(K-K^3)$
F_3	$1.0938(K-K^3)$	$2.392(K-K^3)$	$3.459(K-K^3)$	$1.818(K-K^3)$

$$E_1 = \frac{K}{1 - D_1}.$$

COEFFICIENTS $+ D_1$.

No. of Panel.	B	C	D	E	F	G	H	I	Total.	Values of n.
4	.664	.364	.132						1.160	.4
	.650	.341	.114						1.105	.193
	.644	.335	.110						1.089	.136
	.653	.350	.121						1.124	.25
5	.730	.477	.261	.096					1.564	.4
	.718	.456	.242	.082					1.498	.193
	.713	.450	.233	.075					1.471	.136
	.722	.462	.248	.087					1.519	.25
6	.773	.560	.364	.200	.076				1.973	.4
	.763	.543	.341	.180	.062				1.889	.193
	.760	.538	.335	.173	.058				1.864	.136
	.767	.548	.350	.187	.067				1.919	.25
7	.806	.620	.444	.290	.159	.062			2.381	.4
	.798	.602	.424	.267	.140	.050			2.281	.193
	.794	.597	.417	.260	.133	.045			2.246	.136
	.800	.606	.430	.274	.148	.055			2.313	.25
8	.829	.664	.508	.364	.236	.132	.054		2.787	.4
	.820	.650	.488	.341	.215	.114	.041		2.669	.193
	.820	.644	.482	.335	.208	.110	.038		2.637	.136
	.823	.653	.492	.350	.222	.121	.046		2.707	.25
9	.848	.701	.560	.426	.306	.200	.112	.046	3.199	.4
	.840	.690	.543	.403	.286	.180	.095	.034	3.071	.193
	.838	.684	.538	.398	.277	.173	.090	.032	3.030	.136
	.844	.694	.548	.410	.292	.187	.102	.040	3.117	.25

TABLE I. (*Continued.*)

COEFFICIENTS + D_2.

No. of Panel.	B	C	D	E	F	G	H	I	Total.	Values of n.
4	.243	.391	.342						.976	.4
	.553	.885	.775						2.213	.193
	.800	1.287	1.125						3.212	.136
	.414	.665	.585						1.664	.25
5	.200	.350	.400	.300					1.250	.4
	.455	.792	.905	.680					2.832	.193
	.654	1.153	1.317	.985					4.109	.136
	.340	.597	.682	.512					2.131	.25
6	.170	.308	.391	.385	.263				1.517	.4
	.387	.699	.885	.873	.600				3.444	.193
	.560	1.015	1.287	1.273	.872				5.007	.136
	.286	.527	.665	.659	.452				2.589	.25
7	.145	.272	.365	.402	.363	.233			1.780	.4
	.330	.620	.825	.907	.820	.540			4.042	.193
	.480	.900	1.201	1.320	1.198	.780			5.879	.136
	.246	.465	.620	.683	.620	.400			3.034	.25
8	.130	.243	.335	.391	.396	.342	.212		2.049	.4
	.293	.553	.762	.885	.897	.775	.489		4.654	.193
	.425	.800	1.105	1.287	1.306	1.125	.710		6.758	.136
	.218	.414	.572	.665	.675	.585	.365		3.494	.25
9	.114	.218	.308	.373	.401	.385	.321	.192	2.312	.4
	.261	.498	.699	.842	.906	.873	.727	.440	5.246	.193
	.380	.720	1.015	1.225	1.318	1.273	1.050	.642	7.623	.136
	.194	.372	.527	.634	.681	.659	.550	.330	3.947	.25

COEFFICIENTS + F_2.

No. of Panel.	B	C	D	E	F	G	H	I	Total.	Values of n.
4	.584	1.030	1.212						2.826	.4
	.902	1.540	1.668						4.110	.193
	1.160	1.960	2.026						5.146	.136
	.760	1.314	1.468						3.542	.25
5	.470	.876	1.140	1.206					3.692	.4
	.732	1.330	1.666	1.602					5.330	.193
	.944	1.706	2.090	1.916					6.656	.136
	.620	1.136	1.435	1.430					4.621	.25
6	.398	.752	1.030	1.186	1.191				4.557	.4
	.620	1.152	1.540	1.700	1.540				6.552	.193
	.800	1.480	1.960	2.100	1.817				8.157	.136
	.520	980	1.314	1.475	1.388				5.677	.25
7	.340	.658	.924	1.116	1.206	1.178			5.422	.4
	.530	1.018	1.400	1.640	1.688	1.480			7.756	.193
	.680	1.300	1.788	2.067	2.068	1.732			9.635	.136
	.442	.860	1.190	1.408	1.480	1.350			6.730	.25
8	.300	.584	.830	1.030	1.160	1.212	1.164		6.480	.4
	.470	.902	1.270	1.540	1.686	1.668	1.442		8.978	.193
	.606	1.160	1.625	1.960	2.102	2.026	1.670		11.149	.136
	.396	.760	1.080	1.314	1.458	1.468	1.322		7.798	.25
9	.268	.520	.752	.950	1.100	1.186	1.210	1.150	7.136	.4
	.420	.802	1.152	1.430	1.620	1.700	1.638	1.400	10.162	.193
	.540	1.035	1.480	1.830	2.045	2.100	1.971	1.612	12.613	.136
	.350	.680	.980	1.220	1.387	1.475	1.450	1.295	8.837	.25

TABLE J.

DOUBLE-TRACK DRAW-SPAN. BRACES IN CENTRE PANEL. FOUR SUPPORTS.

COEFFICIENTS — F_3.

No. of Panel.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Total.	Values of *n*.
4	.255	.410	.361						1.026	.4
	.560	.897	.787						2.244	.193
	.810	1.298	1.137						3.245	.136
	.424	.680	.597						1.701	.25
5	.208	.368	.419	.316					1.311	.4
	.455	.806	.918	.690					2.869	.193
	.662	1.163	1.329	.994					4.148	.136
	.348	.611	.698	.522					2.179	.25
6	.176	.324	.410	.403	.280				1.593	.4
	.386	.710	.897	.888	.608				3.489	.193
	.565	1.025	1.298	1.283	.880				5.051	.136
	.297	.540	.680	.670	.462				2.649	.25
7	.152	.287	.382	.420	.381	.247			1.869	.4
	.330	.625	.840	.919	.834	.545			4.093	.193
	.485	.910	1.210	1.328	1.205	.788			5.926	.136
	.252	.477	.636	.700	.632	.410			3.107	.25
8	.135	.255	.353	.410	.415	.361	.224		2.153	.4
	.295	.560	.772	.897	.910	.787	.495		4.716	.193
	.430	.810	1.115	1.298	1.317	1.135	.720		6.825	.136
	.224	.424	.586	.680	.690	.597	.374		3.575	.25
9	.120	.229	.324	.390	.420	.403	.339	.203	2.428	.4
	.261	.500	.710	.856	.917	.888	.738	.446	5.316	.193
	.385	.728	1.025	1.232	1.327	1.283	1.060	.650	7.690	.136
	.200	.380	.540	.649	.698	.670	.560	.338	4.035	.25

TABLE J. (*Continued.*)

COEFFICIENTS — C_3.

No. of Panel.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Total.	Values of *n*.
4	.086	.137	.120						.343	.4
	.098	.158	.140						.396	.193
	.104	.166	.146						.416	.136
	.095	.152	.132						.379	.25
5	.070	.123	.141	.106					.440	.4
	.081	.142	.162	.121					.506	.193
	.086	.149	.170	.128					.533	.136
	.078	.136	.154	.116					.484	.25
6	.060	.109	.137	.134	.093				.533	.4
	.069	.124	.158	.156	.108				.615	.193
	.072	.131	.166	.164	.113				.646	.136
	.066	.120	.152	.149	.103				.590	.25
7	.051	.096	.128	.141	.127	.082			.625	.4
	.059	.110	.148	.162	.148	.096			.723	.193
	.062	.116	.154	.170	.154	.100			.756	.136
	.056	.107	.142	.155	.140	.092			.692	.25
8	.046	.086	.118	.137	.139	.120	.075		.721	.4
	.053	.098	.136	.158	.160	.140	.086		.831	.193
	.056	.104	.143	.166	.169	.146	.091		.876	.136
	.050	.095	.131	.152	.153	.132	.083		.796	.25
9	.041	.077	.109	.131	.140	.134	.112	.068	.812	.4
	.047	.088	.124	.150	.162	.156	.130	.079	.936	.193
	.050	.094	.131	.157	.170	.164	.136	.083	.985	.136
	.044	.085	.120	.144	.155	.149	.124	.078	.899	.25

COEFFICIENTS + C_3.

No. of Panel.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Total.	Values of *n*.
4	.012	.020	.017						.049	.4
	.008	.013	.011						.032	.193
	.006	.010	.009						.025	.136
	.009	.015	.013						.037	.25
5	.010	.018	.020	.015					.063	.4
	.007	.012	.013	.010					.042	.193
	.005	.009	.010	.008					.032	.136
	.008	.014	.016	.012					.050	.25
6	.008	.015	.020	.019	.013				.075	.4
	.006	.010	.013	.013	.009				.051	.193
	.004	.008	.010	.010	.007				.039	.136
	.007	.012	.015	.015	.010				.059	.25
7	.007	.014	.018	.020	.018	.012			.089	.4
	.005	.009	.012	.013	.012	.008			.059	.193
	.004	.007	.009	.010	.009	.006			.045	.136
	.006	.011	.014	.016	.014	.009			.070	.25
8	.006	.012	.017	.020	.020	.017	.010		.102	.4
	.004	.008	.011	.013	.013	.011	.007		.067	.193
	.003	.006	.009	.010	.010	.009	.005		.052	.136
	.005	.009	.013	.015	.015	.013	.008		.078	.25
9	.006	.011	.015	.019	.020	.019	.016	.010	.116	.4
	.004	.007	.010	.012	.013	.013	.010	.006	.075	.193
	.003	.006	.008	.009	.010	.010	.008	.005	.059	.136
	.004	.009	.012	.015	.016	.015	.012	.008	.091	.25

COEFFICIENTS $+ E_1$.

No. of Panel.	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	Total.	Values of *n*.
4	.744	.785	.862						2.391	.4
	.719	.763	.842						2.324	.193
	.708	.750	.836						2.294	.136
	.727	.770	.850						2.347	.25
5	.740	.766	.811	.884					3.201	.4
	.713	.741	.788	.867					3.109	.193
	.703	.730	.777	.863					3.073	.136
	.722	.747	.794	.873					3.136	.25
6	.738	.756	.785	.832	.900				4.011	.4
	.711	.730	.763	.810	.885				3.899	.193
	.700	.719	.750	.800	.882				3.851	.136
	.720	.736	.770	.716	.891				3.833	.25
7	.737	.750	.771	.803	.850	.913			4.824	.4
	.710	.723	.746	.781	.829	.898			4.687	.193
	.699	.713	.735	.770	.822	.894			4.633	.136
	.719	.730	.753	.787	.835	.904			4.728	.25
8	.737	.744	.762	.785	.820	.862	.922		5.632	.4
	.709	.719	.736	.763	.797	.842	.910		5.476	.193
	.698	.708	.725	.750	.786	.836	.904		5.407	.136
	.719	.727	.742	.770	.803	.850	.914		5.525	.25
9	.736	.742	.756	.773	.800	.832	.873	.929	6.441	.4
	.707	.715	.730	.750	.776	.810	.855	.918	6.261	.193
	.697	.705	.719	.738	.765	.800	.850	.913	6.187	.136
	.718	.723	.736	.757	.782	.716	.862	.922	6.216	.25

Positions of Loads to be Considered in Determining the Stresses in a Two-span Draw.

1. The draw swinging, dead load only acting, and all dead load carried to the centre pier.

2. The draw closed and the ends raised, dead load only acting. In this case the ends are raised by wedges, cams, or some similar device, to such an extent that when a full live load is on one arm the other arm will not be lifted off the end support. How much the ends must be raised would be determined by finding the reaction at the end of the unloaded arm when the full load is on the other arm.

3. Each arm considered as an independent span under live load only. The strains in the chords and the web-mem-

bers at the abutment end of the arm are to be found for this loading.

4. Both arms continuous and full live load on each. The chord-stresses to be determined in this case.

5. Both arms continuous, full live load on one arm, and the live load coming on the other arm from the abutment end and advancing by panels until the bridge is fully loaded. The stresses to be determined in the arm carrying the advancing loads for each position of the loads. When the first load is at the abutment the condition is the same as though the ends were latched down and one arm was loaded. The stresses in the unloaded arm for this condition are to be added to the stresses as found for dead load swinging to obtain maximum stresses in parts of the truss.

Positions of Loads to be Considered in Determining the Stresses in a Three-span Draw with the Pier-panel Unbraced.

The positions of the loads giving maximum stresses in this arrangement are the same as for the two-span bridge, the only difference being that the reactions are to be determined by using the tables for three spans with unbraced pier-panel.

Positions of Loads Giving Maximum Stresses in a Three-span Bridge with Bracing in the Centre or Pier Panel.

The conditions are very different from those in the last case, and we have:

1. The draw swinging, dead load only acting.

2. The draw closed and ends raised, dead load only acting (continuous).

3. Full live load on both arms (continuous).

4. As a load on either arm produces a positive reaction at the end of the other arm, and also as the case of one arm as an independent span cannot occur with the pier-panel braced, to obtain the maximum stresses in the members at the shore end of the arm a full load must be placed on the other arm

and the load then advanced by panels from the *centre pier* to the end of the arm whose stresses are being determined.

5. Remove all load from one arm and advance the loads from the *shore end* to the centre pier on the other arm. This will give the maximum stresses in the web-members near centre pier. The stresses in the lower chords near the centre pier will change from plus to minus under the various loadings, the greatest tension being with one arm fully loaded and the panel next the centre pier loaded on the other arm. The stresses in the pier-panel and one arm should be found for full load on one arm and no load on the other. The shear in the pier-panel will be $R_3 + R_4$, noting that the sign of R_3 is minus. If the bracing in the pier-panel is double (see Plate E) each set will take one half the shear. This will make the stresses in the upper and lower portions of the centre posts of different amounts.

The tables give the point at which the moment from a load at any point becomes zero. Thus, in the case we are considering for a load at the third panel in a seven-panel arm, the value of E_1 (see Table H, page 117) is .855; this load will then produce a positive moment—that is, compression in the top chord and tension in the bottom chord—for those chords whose centre of moments lies between the end of the arm and a point .855 from the end. This moment will be negative for any point of moments between .855 and the centre pier.

This furnishes a simple means of determining what loads should be combined to produce the greatest positive or negative moment at any point.

Positions of Loads Giving Maximum Stresses in the Tipper.

The simplest plan is to consider the centre cantilever part of the truss, and the bridge is then treated exactly like the two-span draw, the bracing in the pier-panel being made heavy enough to transfer one half the reaction at this pier from any unequal load—that is, with full load on one arm and none on the other.

DEFLECTION.

Deflection.—It is important that the deflection of the bridge under full load and when swinging be carefully determined, especially if wedges, eccentrics, or some device be used for raising the ends which allows of but a limited vertical movement. To provide against the bridge deflecting below a horizontal line when fully loaded each arm should be given a camber.

Camber.—This camber is arranged for by giving to each member an increase or decrease in length proportional to its stress, the tension-members being shortened and the compression-members lengthened.

The formula for the change in length produced by a given stress is $c = \frac{u \times l}{E}$; $c =$ the change in length, $u =$ the stress per square inch, and $E =$ the modulus of elasticity $=$ for steel 29000000.

Taking for example the bridge whose stresses are given in table on page 126 (see Fig. 60 and details on Plate C), we have for a full dead and live load (span continuous) the sum of stresses in cols. 3 and 8. The areas are given in col. 3 and the resulting stresses per square inch in col. 4. The lengths of the members are given in col. 5 and the values of $\frac{u \times l}{E}$ in col. 6.

Play in Pin-holes.—The lengths are slightly affected by the play in the pin-holes, which, assuming $\frac{1}{32}$ in. as the amount the holes are larger than the pins, will give .0026 in. as the change in length in the end post (0–6) from this cause. The top chord having riveted joints except at 6 and 9, there will be only $\frac{1}{64}$ in. $=$ .0013 in. play at 6 and 9, with none at 7 and 8. In the same way the bottom chord has .0013 at 0 and none at other points. All posts have .0026 and the built diagonals the same amount. The eye-bars are bored $\frac{1}{32}$ in. short, so no allowance will be made for these. Col. 7 gives the change in length from play in pin-holes and col. 8 the total

TABLE OF STRAINS

Mark.	Dead Load.		Single Span.				Live Load Continuous.			
	Dead Load Swinging.	Dead Load Continuous, Upward Reaction at Ends 20000.	As Single Span, Full Load with Excess at 2.	As Single Span, Full Load with Excess at 3.	As Single Span, Loads at 2, 3, and 4 for Strain in 6–2, and Loads at 3 and 4 for Strains in 2–8 and 8–3.		Live Load Continuous, Both Arms Loaded, Excess at 2 and 2.	Live Load Continuous, Both Arms Loaded, Excess at 3 and 3.	Live Load Continuous, One Arm Fully Loaded, Excess at 2 and an Advancing Load at Far End of Other Arm.	Same, with Excess at 3.
1	2	3	4	5	6	7	8	9	10	11
0–6	− 17400	+ 11200	+177000	+173000	....	....	+134000 +124000	+127000 +118000	+151500	+144000
6–7	− 49500	− 9500	+192000	+187500	....	... ...	+129000 +120000	+118000 +109000	+154000	+146000
7–8	− 49500	− 9500	+192000	+187500	... .	....	+129000 +120000	+118000 +109000	+154000	+146000
8–9	−109500	− 50300	+183500	+191500	....	....	+ 90000 + 74000	+ 90000 + 79000	+125000	+133000
9–10	−223500	−150500	——	——	....	....	−107000 −135000	−118000 −134000	− 68000	− 68000
10–10	−215500	−144500	——	——	... ...	....	−102000 −130000	−113000 −128000	− 65000	− 65000
0–1	+ 13000	− 8200	−126500	−125000	....	....	− 96000 − 89000	− 92000 − 85000	−107500	−103000
1–2	+ 13000	− 8200	−126500	−125000	....	....	− 96000 − 89000	− 92000 − 85000	−107500	−103000
2–3	+108600	+ 50000	−180000	−190000	....	....	− 89000 − 73500	− 89000 − 77500	−124000	−132500
3–4	+172800	+101500	−107000	−110000	... .	....	+ 3000 + 23000	+ 7000 + 22500	− 40000	− 43000
4–5	+172800	+101500	−107000	−110000	... .	....	+ 3000 + 23000	+ 7000 + 22500	− 40000	− 43000
5–5	+215500	+144500	——	——	....	....	+102000 +130000	+113000 +128000	+ 65000	+ 65000
1–6	− 16400	− 16400	− 56000	− 56000	....	76000	− 56000 − 56000	− 56000 − 56000	− 56000	− 56000
6–2	+ 50500	+ 24000	− 90000	− 86000	− 73000 −108500	....	− 47000 − 42500	− 36000 − 34500	− 53500	− 60000
2–7	+ 8200	+ 8200	——	——	....	... ...	——	——	——	——
2–8	− 84000	− 58000	− 17000	+ 5000	....	+ 29000 + 38500	− 58000 − 66500	− 42000 − 45000	− 43000	− 20000
8–3	+ 56300	+ 43500	+ 30500	+ 15000	... ...	− 14000 − 28500	+ 48000 + 55000	+ 34000 + 40000	+ 42500	+ 28000
3–9	− 96800	− 79200	−110000	−121000	....	....	−139000 −147000	−146000 −153000	−130000	−137000
9–4	− 16400	− 16400	− 56000	− 56000	....	76000	− 56000 − 56000	− 56000 − 56000	− 56000	− 56000
9–5	+ 65000	+ 65500	+162000	+166000	....	....	+156000 +159500	+162000 +164000	+160000	+164500
5–10	+ 75700	+ 49700	——	——	....	....	+ 29000 + 38000	+ 31000 + 37000	+ 19000	+ 18500

IN 293-FOOT SPAN.

Live Load Continuous.				Combinations.				
Strains in Unloaded Arm, Other Arm Fully Loaded.	Full Load on One Arm, Advancing Load at 1 on Other Arm.	Full Load on One Arm, Advancing Load at 2 on Other Arm.	Full Load on One Arm, Advancing Load at 3 on Other Arm.	Dead Load Swinging, Live Load as Single Span.	Dead Load Swinging, Live Load Continuous (Case 12).	Dead Load Swinging, Live Load Continuous (Case 8 or 9)	Dead Load Continuous, Live Load Continuous, in Various Positions.	Totals.
12	13	14	15	16	17	18	19	
+ 3000 − 26500	+ 55500	+ 90000	+108500	+159600	− 43900	....	+162700	− 43900 +162700
+ 4000 − 37000	+ 3000	+ 72000	+ 96500	+142500	− 86500	....	− 46500 +144500	− 86500 +144500
+ 4000 − 37000	+ 3000	+ 72000	+ 96500	+142500	− 86500	....	− 46500 +144500	− 86500 +144500
+ 6000 − 54500	− 35000	+ 4000	+ 57500	+ 82000	−164000	....	−104800 + 82700	−104800 + 82700
+ 7000 − 67000	− 79500	− 96000	−121500	−223500	−290500	−358500	−285500	−358500
+ 6800 − 64000	− 76500	− 92500	−117000	−215500	−279500	−345500	−274500	−345500
− 2000 + 19000	− 40000	− 64500	− 78000	−113500	+ 32000	....	−115700	−115700 + 32000
− 2000 + 19000	− 40000	− 64500	− 78000	−113500	+ 32000	....	−115700	−115700 + 32000
− 5600 + 54000	+ 35000	− 4000	− 58000	− 81400	+162600	....	− 82500 + 85000	− 82500 +162600
− 7000 + 64000	+ 63500	+ 55000	+ 46000	+ 65800	+236800	....	+165000	+236800
− 7000 + 64000	+ 63500	+ 55000	+ 46000	+ 65800	+236800	....	+165000	+236800
+ 64000	+ 76500	+ 92500	+117000	+215500	+279500	+345500	+274500	+345500
——	− 76000	− 56000	− 56000	− 92400	− 16400	....	− 92400	− 92400
− 2800 + 25000	+ 53000	− 9500	− 26000	− 58000	+ 75500	....	− 36000 + 77000	− 36000 + 77000
——	——	——	——	+ 8200	+ 8200	....	+ 82000	+ 8200
+ 2500 − 24500	.... − 62000	− 86000 − 97000	.... − 54500	−101000 + 38500	−108500	−150500	−155000	−155000 + 38500
− 1400 + 11500	.... + 33000	+ 63000 + 69000	.... + 45000	− 28500 + 86000	+ 67800	+111300	+102500	− 28500 +111300
+ 1500 − 15500	.... − 44000	.... − 92000	−151000 −159000	−217800	−112300	−249800	−238200	−249800
——	——	——	——	− 92400	− 16400	....	− 92400	− 92400
——	+ 20000	+ 55000	+107000	+231000	+ 65000	+229000	+230000	+231000
+ 18700	+ 22010	+ 28000	+ 33500	+ 75700	+ 94400	+113700	+ 87700	+113700

NOTE.—Upper set of figures in columns 8 to 15 give strains with bracing in centre panel; strain in (9–5) full load on one arm, other arm unloaded, in case of bracing in centre panel = 171000 against 164000 with bracing omitted (see col. 9).
Strain in lower half of (10–5) = 294000, and in the diagonal in centre panel 166000 if double set are used, and 305000 if one set is used.
NOTE.—Upper figures in columns 6–7 are for the condition of bracing in centre panel making spans continuous. The strain in the end post (0–6) is 137000 against 177000 in col. 4.

change in length. These values are given for convenience the same sign as the strain in the member. The change in elevation of any point is now found by placing a load of one (1) at the point in question and finding the stresses in the various members for this load of one, and multiplying these stresses by the values already determined for the changes in length of the several members (col. 8). Col. 9 gives the stresses for a load of one at 3, and col. 10 gives the results of col. 8 multiplied by col. 9. Adding col. 10 algebraically, we have .05475 in. = $\frac{21}{32}$ in. as the deflection at 3. In the same manner the deflection at any other point is found. We wish particularly the change at the end (at 0) due to the changes caused by camber,* and, placing a load of one at the end and finding the stresses (called coefficients) for this load, we have col. 11. Multiplying cols. 8 and 11, the results in col. 12 are obtained, which, added algebraically, give .08073 in. = $\frac{31}{32}$ in. as the deflection at the end of the arm due to the changes in the lengths of the various members which were made to produce the camber.

Deflection from Dead Load Swinging.—Proceeding in a manner similar to the above, we have in Table D_2 (page 132), col. 2, the stresses from dead load swinging (taken from table on page 126). In col. 3 are the areas, in col. 4 the unit-stresses, in col. 5 the lengths, in col. 6 the change in length from these stresses, in col. 7 the play in the pin-holes, in col. 8 the total changes, in col. 9 the coefficients from a load of one at the end, and in col. 10 the effect at the end produced by these changes in the several members. The results in col. 10 added algebraically give .23254 ft. = $2\frac{13}{16}$ in. as the deflection at the end of the arms when the draw is swinging produced by the dead load.

Adding the deflection produced by the camber and by the dead load, we have

$$\frac{31}{32} \text{ in.} + 2\frac{13}{16} \text{ in.} = 3\frac{25}{32} \text{ in.} = \text{total deflection.}$$

* The changes in length for camber are the changes produced by a full load on both arms, considering the bridge as a continuous girder.

A portion of this deflection we wish to take up by driving the end wedges and the remainder is to be taken up by shortening the top chord (9–10). Any change in the length of (9–10) produces a change at the end in proportion as the length of the arm is to a perpendicular letf all from (9–10) to the point (5).*

Amount of Deflection Taken Up by the End Wedges.—When bracing is used in the centre panel of a three-span draw there can be no negative reaction at the ends from a partial load, and hence the wedges need only be driven to a bearing. In a three-span bridge with the centre panel unbraced or in a two-span bridge there will be a negative reaction under a partial load, and the wedges should lift the ends enough to at least overcome any negative reaction. Loading one arm and finding the reaction at the end of the unloaded arm will then determine the upward force the wedges must exert.

Assume that it has been found that the negative reaction never exceeds 20000 lbs., and that this is the force it has been decided to have the wedges exert. We now place a load of 20000 lbs. *acting upward* at the end of the arm and find the stresses in the various members for this load. Col. 2, Table D_3, gives these stresses, col. 3 the areas, col. 4 the unit-stresses, col. 6 the change in length, col. 7 the coefficients for a load of one (as previously determined), and col. 8 the resulting deflection at the end $= .08609$ in. $= 1\frac{1}{32}$ in., which is the amount the ends will be lifted by the wedges. $3\frac{25}{32} - 1\frac{1}{32} = 2\frac{3}{4}$ in. as the deflection which must be taken up by shortening top chord (9–10).

Deflection due to Changes in Temperature.—In a city bridge the lower chords are often covered over by the floor or perhaps entirely closed in. In this case, being protected from the sun, the chords may have a temperature of 30° or more lower than the other parts of the truss which are exposed to

* The change is also as the ratio of the coefficient of (9–10) for a load of one (1) is to one (1).

the direct action of the sun. It is sometimes important to know just what deflection will be produced by this condition.

The change of length produced by a change of one degree Fahrenheit in temperature is, say, .0000063 per cent of the length; for 30° this change = .000019 per cent. Multiplying the lengths of the members affected by the sun by .000019, we have the results given in col. 2 of Table D_4. These are now multiplied by the coefficients for a load of one at the end, and the amounts are given in col. 3, which being added algebraically give .068 ft. = $\frac{13}{16}$ as the deflection from this cause. The wedges should have throw enough to raise the end this additional amount, or $1\frac{1}{32} + \frac{13}{16} = 1\frac{27}{32}$ in. The power necessary to lift the ends this amount is also increased about 80 per cent. The neglect of the deflection caused by temperature-stresses is no doubt one cause which has led to the unsatisfactory working of wedge-lifts, and the consequent disfavor with which they are regarded by some engineers.

Cause of Failure of Wedge-lifts.—Another cause which has been a common source of trouble with wedge machinery has been the use of too small shafting. Where the shafts run the full length of the bridge and drive four end wedges, and perhaps two centre ones as well, the torsion in the long shafts may easily be enough, and of such unequal amounts, as to render the driving of all the wedges equally, practically impossible. The multiplication of the power should of course be made at the ends and not at the centre, reducing the stress on the shafting as much as possible. This trouble with the shafting was probably one reason for the practice, which is now common, of driving the end machinery by a double set of rods instead of shafts (see Plates A and B). The rod arrangement is also cheaper and simpler; the use of small gears, which has been a fruitful source of breakage, is avoided. Where other power than hand-power is used to operate the draw the rod arrangement is not applicable, and its use is therefore limited to short or light spans.

Table D₁.—DEFLECTION UNDER FULL DEAD AND LIVE LOADS.

Mark	Strain (Cols. 3 and 5, Table C).	Area.	Unit-stress.	Length.	Change in Length.	Play in Pin-holes.	Totals.	Coeff. for Ld. at Panel-point No. 3.	Deflection at No. 3	Deflection at End Due to Change of Lengths for Camber.	
1	2	3	4	5	6	7	8	9	10	11	12
0–6	+135.2	23.5	+ 5750	38.78	$\frac{5750 \times 38.78}{29000000} = + .007685$	+.0026	+.010285	+ .335	+.00344	+ .01285 × —1.44 =	—.01481
6–7	+110.0	19.4	+ 5670	27.84	$\frac{5670 \times 27.84}{29000000} = + .005444$	+.0013	+.006744	+ .465	+.00313	+.006744 × —2.00 =	—.01348
7–8	+110.0	19.4	+ 5670	27.84	" " "	"	"	+ .465	+.00313	+.006744 × —2.00 =	—.01348
8–9	+ 24.0	19.4	+ 1230	28.04	$\frac{1230 \times 28.84}{29000000} = + .001189$	+.0026	+.003789	+ .705	+.00267	+.003789 × —2.95 =	—.01117
9–10	—285.0	27.0	—10500	28.84	$\frac{10500 \times 28.84}{29000000} = - .010443$	....	—.010443	— .550	+.00574	—.010443 × —3.67 =	+.03832
10–10	—274.0	25.5	—10750	7.5	$\frac{10500 \times 7.5}{29000000} = - .005431$	......	—.005431	— .268	+.00145	—.005431 × —3.55 =	+.01928
0–1	— 97.2	19.4	— 5010	27.81	$\frac{5010 \times 27.81}{29000000} = - .004806$	—.0013	—.006106	— .235	+.00143	—.006106 × +1.04 =	—.00635
1–2	— 97.2	19.4	— 5010	"	" " "	....	—.004806	— .235	+.00112	—.004806 × +1.04 =	—.00499
2–3	— 23.5	19.4	— 1211	"	$\frac{1211 \times 27.81}{29000000} = - .001162$	....	—.001162	— .700	+.00077	—.001162 × +2.91 =	—.00338
3–4	+124.5	19.4	+ 6420	"	$\frac{6420 \times 27.81}{29000000} = + .006157$	....	+.006157	... .	....	+.006157 × +3.54 =	+.02179
4–5	+124.5	19.4	+ 6420	"	" " "	....	+.006157	....	....	+.006157 × +3.54 =	+.02179
5–5	+274.5	29.4	+ 9340	7.5	$\frac{9340 \times 7.5}{29000000} = + .004831$	......	+.004831	+ .268	+.00129	+.004831 × +3.54 =	+.01701
6–2	— 18.5	14.7	— 1250	38.75	$\frac{1250 \times 38.75}{29000000} = - .001672$	—.0026	—.004272	— .335	+.00143	+.004272 × +1.35 =	—.00576
2–8	—124.5	11.25	—11000	39.81	$\frac{11000 \times 39.81}{29000000} = - .001510$	.. ...	—.001510	+ .325	—.00049	—.001510 × —1.31 =	+.00198
8–3	+ 98.5	14.7	+ 6700	32.0	$\frac{6700 \times 32.0}{29000000} = + .007393$	+.0026	+.009993	— .160	—.00155	+.009993 × + .70 =	+.00699
3–9	—226.2	19.5	—11600	42.38	$\frac{11600 \times 42.38}{29000000} = - .016051$	....	—.016951	—1.085	+.01839	—.016951 × — .93 =	+.01577
9–5	+225.0	29.4	+ 7650	42.38	$\frac{7650 \times 42.38}{29000000} = + .011180$	+.0026	+.013780	+ .805	+.01110		
10–5	+ 87.7	13.7	+ 6410	40.00	$\frac{6410 \times 40.0}{29000000} = + .008840$	+.0026	+.011440	+ .150	+.00171	+.011440 × + 1.0 =	+.01144
									+.05475'' = $\frac{3}{32}$''		+.08104'' = $\frac{3}{32}$''

TABLE D_2.—DEFLECTION WITH DEAD LOAD SWINGING.

Mark	Strain.	Area.	Unit-stress.	Length.	Change in Length.	Play in Pin-holes.	Total Change.	Coefficient for Load at End.	Deflection at End.
1	2	3	4	5	6	7	8	9	10
0-6	− 17.400	23.5	−.700	38.78	$\frac{-740 \times 38.78}{29000000} = -.000989$	−.0026	−.003589	−1.44	+.00517
6-7	− 49.5	19.4	−2551	27.84	$\frac{-2551 \times 27.84}{29000000} = -.002448$	−.0013	−.003748	−2.0	+.00749
7-8	− 49.5	19.4	−2551	27.84	$\frac{-2551 \times 27.84}{29000000} = -.002448$	−.0013	−.003748	−2.0	+.00749
8-9	−109.5	19.4	−5644	28.04	$\frac{-5644 \times 28.04}{29000000} = -.005457$	−.0026	−.008057	−2.95	+.02380
9-10	−223.5	27.0	−8278	28.84	$\frac{-8278 \times 28.84}{29000000} = -.008232$	....	−.008232	−3.67	+.03021
10-10	−215.5	25.5	−8450	7.5	$\frac{-8450 \times 7.5}{29000000} = -.004371$	....	−.004371	−3.55	+.01551
0-1	+ 13.0	19.4	+.670	27.81	$\frac{+670 \times 27.81}{29000000} = -.000642$	+.0013	+.001942	+1.04	+.00202
1-2	+ 13.0	19.4	+.670	27.81	" "	....	+.000642	+1.05	+.00677
2-3	+108.6	19.4	+5598	27.81	$\frac{+5598 \times 27.81}{29000000} = +.005368$	....	+.005368	+2.91	+.01562
3-4	+172.8	19.4	+8908	27.81	$\frac{+8908 \times 27.81}{29000000} = +.008544$	....	+.008544	+3.54	+.03024
4-5	+172.8	19.4	+8908	27.81	" "	....	+.008544	+3.54	+.03024
5-5	+215.5	29.4	+7330	7.5	$\frac{+7330 \times 7.5}{29000000} = +.003791$	....	+.003791	+3.54	+.01342
6-2	+ 50.5	14.7	+3430	38.75	$\frac{+3430 \times 38.75}{29000000} = +.004583$	+.0026	+.007183	+1.35	+.00969
2-8	− 84.0	11.25	−7460	39.81	$\frac{-7460 \times 39.81}{29000000} = -.010240$	....	−.010240	−1.31	+.01341
8-3	+ 56.3	14.7	+3830	32.0	$\frac{+3830 \times 32.0}{29000000} = +.004226$	+.0026	+.006826	+ .70	+.00477
3-9	− 96.8	19.5	−4960	42.38	$\frac{-4960 \times 42.38}{29000000} = -.007249$	....	−.007249	− .93	+.00674
9-5	+ 65.0	29.4	+2210	42.38	$\frac{+2210 \times 42.38}{29000000} = +.003230$	+.0026	+.005830	——	——
10-5	+ 75.7	13.7	+5530	40.0	$\frac{+5530 \times 40.0}{29000000} = +.007351$	+.0026	+.009951	+1.00	+.00995
									+.23254 = $2\frac{13}{16}$

TABLE D_3.

DEFLECTION. UPWARD REACTION ONLY AT END. 20,000 LBS.

Mark.	Strain.	Area.	Unit Stress.	Length.	Change in Length.	Coeff. for Load of One at End.	Deflection at End.
1	2	3	4	5	6	7	8
0–6	+ 28.8	23.5	1225	38.78	$\frac{1225 \times 38.78}{29000000} = +.00163$	− 1.44	− .00234
6–7	+ 40.0	19.4	2060	27.84	$\frac{2060 \times 27.84}{29000000} = +.00197$	− 2.0	− .00394
7–8	+ 40.0	19.4	2060	"	" "	− 2.0	− .00394
8–9	+ 59.0	19.4	3040	28.04	$\frac{3040 \times 28.04}{29000000} = +.00293$	− 2.95	− .00864
9–10	+ 73.4	27.0	2704	28.84	$\frac{2704 \times 28.84}{29000000} = +.00268$	− 3.67	− .00983
10–10	+ 71.0	25.5	2780	7.5	$\frac{2780 \times 7.5}{29000000} = +.00072$	− 3.55	− .00255
0–1	− 20.0	19.4	1030	27.81	$\frac{1030 \times 27.81}{29000000} = -.00097$	+ 1.04	− .00100
1–2	− 20.0	19.4	1030	"	" "	+ 1.04	− .00100
2–3	− 58.2	19.4	3000	"	$\frac{3000 \times 27.81}{29000000} = -.00287$	+ 2.91	− .00834
3–4	− 70.8	19.4	3613	"	$\frac{3613 \times 27.81}{29000000} = -.00346$	+ 3.54	− .01210
4–5	− 70.8	19.4	3613	"	" "	+ 3.54	− .01210
5–5	− 70.8	29.4	2408	7.5	$\frac{2408 \times 7.5}{29000000} = -.00062$	+ 3.54	− .00219
6–2	− 27.0	14.7	1837	38.75	$\frac{1837 \times 38.75}{29000000} = -.00245$	+ 1.35	− .00329
2–8	+ 26.2	11.25	2329	39.81	$\frac{2329 \times 39.81}{29000000} = +.00319$	− 1.31	− .00418
8–3	− 14.0	14.7	952	32.0	$\frac{952 \times 32}{29000000} = -.00105$	+ .70	− .00735
3–9	+ 18.6	19.5	954	42.38	$\frac{954 \times 42.38}{29000000} = +.00139$	− .98	− .00129
9–5	....	29.4	....	"	....	....	... .
10–5	− 20.0	13.7	1462	40.0	$\frac{1462 \times 40.00}{29000000} = -.00201$	+ 1.0	− .00201
							− .08609 = $1\frac{1}{32}''$

TABLE D_4.

TEMPERATURE.

Length Multipl'd by .000019.	Results of col. 6, Table D_3 × col. 2, Table D_4.
2	3
+ .0078	− .0105
+ .0053	− .0106
+ .0053	− .0106
+ .0054	− .0160
+ .0055	− .0205
+ .0014	− .0049
+ .0073	+ .0098
+ .0076	− .0100
+ .0061	+ .0043
+ .0080	− .0074
+	....
+ .0076	+ .0076
	− .0688 = $\frac{13}{16}''$

THE WORK TO BE PERFORMED OR THE FORCES TO BE OVERCOME IN THE OPERATION OF A DRAWBRIDGE.

1. The first force to be considered will be that which is necessary to lift the ends the desired amount when the span is closed, that is, the force required to drive the end wedges, including the friction of the machinery, etc.

2. The force required to overcome the inertia of the bridge. A mass represented by the weight of the span must be moved through an arc of 90° when the bridge is opened or closed. This mass will not be moved with a uniform velocity, but during a portion of the movement the velocity will be accelerated, during another portion the velocity will be retarded, and there may or may not be a period during which a constant velocity is maintained.

3. The force necessary to overcome the friction of the various moving parts of the machinery, etc.

4. The force necessary to overcome the effect produced by the wind when assumed as acting on one arm only of the bridge. This force can only be assumed, and many engineers make provision for a wind-force sufficiently great that if the machinery be designed to overcome this wind-force it will be ample to operate the draw under ordinary conditions and the frictional forces may be neglected in the computation.

Power for Lifting the Ends.—*Constant Velocity.*—As already stated (see page 93), in a two span draw or a three-span draw without bracing in the centre panel the ends must either be latched down or they must be raised to such an extent that when one arm is fully loaded and the other arm is without load the end of the unloaded arm will not rise off its support. The second plan is the more feasible and is the one usually adopted.

In lifting the ends the resistance is zero at the start and increases uniformly to a maximum at the end of the lift. If

the velocity of the moving power be uniform, however, the resistance is moved equal spaces in equal intervals of time, and the power must be equal throughout the movement to the maximum demand upon it at the end, although the full power is not required during the first part of the movement. This is equivalent to lifting the maximum weight, or resistance, through a height h in the time t. If W represent the maximum resistance and P the power required, we have the general equation $P = \frac{Wh}{t}$. P represents the rate at which the work is being done in ft.-lbs. per second. Taking the bridge which we have been considering, we found that a force of 20000 lbs. must be exerted at each wedge or under the ends of each truss (see page 129). This does not include the friction of the wedges, shafting, etc., and for preliminary computations we will increase the resistance 50 per cent, giving us $20000 + 10000 = 30000$ lbs. at each wedge, or 120000 lbs. for the four wedges as the total resistance to be overcome. We found (page 129) that the ends must be raised $1\frac{1}{32}$ in. $= .086$ ft. to give a reaction of 20000 lbs. The work to be done is then $120000 \times .086$ ft. $= 10320$ ft.-lbs. If it is decided to lift the ends in 30 seconds we have $10320 \div 30 = 344$ ft.-lbs. per second as the power. One horse-power equals 550 ft.-lbs. per second, and $344 \div 550 = .63$ H. P. required. Suppose the draw is to be operated by two men, who it is assumed can exert a push of 75 lbs. each against the hand-lever. The power must then be multiplied by the wedges and gearing and hand-lever $120000 \div 150 = 800$ times. The space they must move over $= 120000 \times .086 \div 150 = 68.8$ ft. They can walk at an average rate of, say, three (3) ft. per second; then $68.8 \div 3 = 22.9$ seconds is the time required for them to lift the ends.

Variable Velocity.—If the velocity of the power can be variable the amount of the power will be reduced. The average resistance is 120000 lbs. $\div 2 = 60000$ lbs., and the

work performed is $60000 \times .086 = 5160$ ft.-lbs. Suppose that as the resistance increases the velocity decreases until at the end of the lift the velocity becomes 0. If h and t remain as before $P = \frac{Wh}{2t}$, one half what it was in the case of uniform velocity. If the time and the space (S) passed over by the power remain as above the average velocity equals $S \div t$, which is the same as the uniform velocity in the first case. The maximum velocity will be twice the average $= 2S \div t$.

Suppose, again, that v represents the velocity at the end of the lift, which is to be any assumed amount; that v_b represents the velocity at the beginning of the lift or the initial velocity of the moving power; $a =$ the $\left\{ \begin{matrix} \text{retardation or} \\ \text{acceleration} \end{matrix} \right\}$ of the power or the $\left\{ \begin{matrix} \text{decrease or} \\ \text{increase} \end{matrix} \right\}$ in its velocity per second: then we have from the general equation

$$S = v_f t + \frac{at^2}{2}, \quad a = \frac{2S - 2v_f t}{t^2}.$$

If the time be assumed as 30 seconds, S to be 68.8 ft., and the final velocity be made 1.5 ft. per second, then

$$a = \frac{(2 \times 68.8) - (2 \times 1.5 \times 30)}{900} = .053;$$

the initial velocity

$$= v_f + at = 1.5 + .053 \times 30 = 3.1 \text{ ft. per sec.}$$

Power to Overcome Inertia.—It is necessary first to determine the moment of inertia of the bridge about the axis of rotation, the centre pivot. It is nearly enough correct to assume the draw as a rectangular parallelopipedon of uniform density, whose length and breadth are the same as those of the bridge and whose depth parallel to the axis of rotation is one (1), the weight being assumed as uniform and equal to the weight of the bridge above the drum, the

drum, centre girders, etc., being considered separately. The formula for the moment of inertia of the portion above the drum is

$$I = \frac{W[(\frac{1}{2}l)^2 + (\frac{1}{2}w^2)]}{96.6},$$

given in terms of the mass, or of the weight divided by 32.2. l = the length and w = the width of the bridge. It is convenient to replace the mass of the bridge by a mass at the rack-circle which will produce the same moment of inertia. This mass will be the moment of inertia of the bridge divided by the square of the radius of the rack-circle, or the equivalent mass at rack is

$$M_T = \frac{I}{R^2}.$$

Letting a represent the half-length of the bridge, b the half-width of the bridge, R the radius of the rack-circle, W the weight of the bridge above the drum, and M_T the equivalent mass at rack-circle, then

$$M_T = \frac{W(a^2 + b^2)}{96.6R^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

For determining the moment of inertia of the drum, centre girders, etc., the weight of these parts will be considered as assembled at the drum and the moment of inertia will be $\frac{W_1R_1^2}{32.2}$. R_1 = the radius of the drum and W_1 = the weight of drum, etc.

The equivalent mass at rack-circle is

$$M_D = \frac{W_1R_1^2}{32.2R^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The total mass whose inertia is to be overcome and which is to be moved through an arc of 90° in opening and closing the draw is then

$$M = M_T + M_D. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If it is desired to determine the moment of inertia more carefully than by the above method the moments of inertia of the several parts, as the floor-beams, stringers, etc., may be determined separately. The weights of the truss-members may be taken by panels, and the moments of inertia of each about the axis of rotation may be found and the sum of all the partial moments then taken. The moment of inertia of each part about the centre pivot or axis of rotation is equal to its moment of inertia about an axis through its own centre of gravity, parallel to the axis of rotation, plus the product of its weight or mass by the square of the distance of its centre of gravity from the axis of rotation (see any text-book on mechanics).

The following example of the Thames River Bridge,* which is figured from actual weights, well illustrates the method, and also shows how closely the approximate rule given above checks with the results obtained by using the much more tedious method of parts. In the example which follows I is found in terms of the weight and can be reduced to terms of mass by dividing by 32.2.

The various steps in the process are as follows:

1. *I for the truss proper.* The weights are reduced to loads at the joints, and these joint loads are multiplied by the square of the distances from the joints to the centre of the middle panel or the axis of the truss. To these results is added the product of the weight of the truss by square of the distance between the axis of the truss and the axis of the bridge (this distance is half the width of the bridge centre to centre of trusses).

2. *I of the floor-beams.* The moment of inertia is first found for an axis through the centre of gravity of a beam parallel to the axis of rotation and to this is added the product of the weight of the beam by the square of the dis-

* See Trans. Am. Soc. C. E., Dec. 1891.

tance from its centre of gravity to the axis of rotation. This is repeated for each beam.

3. *I for the stringers, rails, and guard-rails.* The whole length of the bridge can be included in one operation for these parts. I is first found for an axis through the centre of the web in the centre panel, and to this result is added the product of the weight of the line of stringers, etc., by the square of the distance from the centre of the web to the axis of rotation. This is repeated for each line of stringers.

4. *I of the timber floor.* This is considered as a flat rectangular strip of uniform weight per cubic foot, and the moment of inertia $= \frac{W(a^2 + b^2)}{3}$. The length of the truss is 497.48 ft. and the width 28.32 ft. centre to centre. In the table below the values of I are given for one half of one truss and for one half the span for the floor-beams.

For simplicity each floor-beam is considered a parallelopipedon 27 ft. $\times$ 4 ft. $\times$ .166 ft., and $W = 8830$.

$$I = \tfrac{1}{8} \times 8830 \times 182.257 = 536440$$

for one beam about its own centre of gravity. There are 22 floor-beams.

For the stringers we have a parallelopipedon 500 ft. $\times$ 2.5 ft. $\times$ .166 ft., and $W = 90533$ for one line.

$$I = \tfrac{1}{8} \times 90533 \times 62501.563 = 1886170000.$$

There are four lines of stringers distant three (3) and nine (9) feet from the axis of rotation.

For the timber floor we have a parallelopipedon 500 ft. $\times$ 18 ft. $\times$ .75 ft., and $W = 206667$ lbs.

$$I = \tfrac{1}{8} \times 206667 \times 62581 = 4271000000.$$

The table below gives the results of the several computations:

TABLE K.

MOMENT OF INERTIA OF 500-FOOT SPAN.

Penal-point.	Distance from Centre of Bridge.	Square of Distance.	Truss.		Floor-beams.	
			Truss Weights.	Products of quantities in Col. 3 multiplied by those in Col. 4.	Weight of Floor-beams.	Products of quantities in Col. 3 multiplied by those in Col. 6.
1	2	3	4	5	6	7
0	248.74	61,851.69	17,700	1,094,775,090	13,400	828,812,780
1	224.91	50,580.01	26,410	1,335,817,800	8,830	446,621,400
2	201.07	40,441.21	25,660	1,037,721,192	8,830	357,095,796
3	177.24	31.399.84	29,660	931,318,068	8,830	277,260,234
4	153.41	23,381.56	30,160	705,189,056	8,830	206,459.528
5	129.57	16,796.16	33,660	565,360,092	8,830	148,310,446
6	105.74	11,172.49	38,160	426,342,600	8,830	98,653,175
7	81.91	6,707.61	34,660	232,485,416	8,830	59,228,108
8	58.07	3,375.61	39,660	133,876,296	8,830	29,806,548
9	34.24	1,169.64	38,660	45,216,736	8,830	10,327,568
10	10.41	108.16	92,660	10,025,812	8,830	955,406
		Sum	407,050	6,518,128,158		2,463,530,989
				Mult. by 4		Mult by 2
				26,072,512,632		4,927,061,978

TOTAL AMOUNTS.

1. Truss,	$6,518,128,158 \times 4$	$= 26,072,512,632$
2. "	$407,050 \times (14.6)^2 \times 4 =$	326,454,100
3. Floor-beams,	$2,463,530,989 \times 2 =$	4,927,061,978
4. " "	$536,440 \times 22 =$	11,801,680
5. Stringers,	$1,886,100,000 \times 4 =$	7,554,400,000
6. "	$90,533 \times 3^2 \times 2 =$	1,629,594
7. "	$90,533 \times 9^2 \times 2 =$	14,666,364
8. Wood floor,		4,271,000,000
	Total, $I =$	43,279,526,348

WEIGHTS.

Truss, drum, etc.,	$407,050 \times 4 =$	1,628,200
Floor-beams,	$8830 \times 20 + 13,400 \times 2 =$	203,400
Stringers, rails, etc.,	$90,533 \times 4 =$	362,132
Timber,		206,667
Total.................		2,400,399

$$\text{Radius of gyration, } r^2 = \frac{I}{W} = \frac{43,279,526,348}{2,400,399} = 18,030.$$

$$r = 134.3'.$$

In the preceding computations the weight of the drum, etc., has been included with the weight of the trusses. For the purpose of comparison we will now find the moment of inertia of the same bridge by means of the formulæ given above. $W =$ the weight of the trusses, flooring, etc., $= 2185000$ lbs., and $W_1 =$ the weight of the drum and the parts considered with it $= 215000$ lbs.

Repeating the formulæ:

$$I \text{ for the trusses, etc., in terms of the weight} = \frac{W(a^2 + b^2)}{3}, \quad (1)$$

$$I \text{ " " " " " " " " mass} = \frac{W(a^2 + b^2)}{96.6}, \quad (2)$$

$$I \text{ " " drum, etc., " " " " weight} = W_1R_1^2, \quad . \quad (3)$$

$$I \text{ " " " " " " " " mass} = \frac{W_1R_1^2}{32.2}. \quad . \quad (4)$$

$$\text{Total: } I = \frac{W(a^2 + b^2)}{3} + W_1R_1^2 \quad . \quad . \quad . \quad (5)$$

or

$$I = \frac{W(a^2 + b^2)}{96.6} + \frac{W_1R_1^2}{32.2}. \quad . \quad . \quad . \quad (6)$$

$$I = 2185000 \times (250^2 + 14^2) + 215000 \times 16^2 = 45717170000,$$

against 43279526000 by the first process, a difference of about 5 per cent.

Taking the 290-ft. span (Plates C, D, and E), the weights are: trusses, etc., 400000 lbs.; drum, etc., 23000 lbs.; machinery, 55000 lbs.; wood floor, etc., 116000 lbs.; total, 594-000 lbs. Including with the drum 30000 lbs. of machinery, we have $W = 400000 + 25000 + 116000 = 541000$ lbs., and $W_1 = 23000 + 30000 = 53000$ lbs.

$$I = \frac{541000(145^2 + 8^2)}{3} + 53000 \times 9^2 = 3807360700.$$

The radius of the rack-circle is 9.8 ft. and the equivalent mass at the rack is then

$$M = \frac{3807360700}{32.2 \times (9.8)^2} = 1231130 \text{ lbs.}$$

Accelerating Force. — If F_1 = the force required to accelerate the draw or overcome its inertia, f = the acceleration at the rack, and M = the mass at rack, then $F_1 = Mf$.

Acceleration.—The motion of the draw may be accelerated for the first half of the swing and retarded for the last half; or it may be accelerated until a given velocity is attained, this velocity be maintained for a period, and then the motion be retarded until the bridge is brought to rest.

Let R_1 = the radius of the drum in feet = $\frac{1}{2}D$;

R = the radius of the rack;

f = the acceleration in feet per second at the rack-circle $= \frac{v}{t} = \frac{2s_1}{t^2}$;

r = the retardation in feet per second at the rack-circle;

v = the maximum velocity in feet per second at the rack-circle;

T = the total time in seconds required to open the bridge;

v_a = the maximum angular velocity $= \frac{v}{R}$;

g = constant for gravity = 32.2;

S = one quarter the circumference of the rack-circle $= \frac{2\pi R}{4}$;

s_1 = the space on rack in which the motion is to be accelerated;

s_2 = the space on rack in which the motion is to be retarded;

$S-(s_1+s_2)=$ the space on rack in which the motion is to be uniform;
$t=$ time of acceleration;
$t_1=$ time of uniform velocity;
$t_2=$ time of retardation;
$F_1=$ accelerating force.

Then $T=t+t_1+t_2$.

Case (*1*). We have then when the velocity is to be accelerated during half the turning

$$f=\frac{2s_1}{t^2}=\frac{S}{t^2}=\frac{2\pi R}{4t^2}; \qquad F_1=Mf=\frac{M\pi R}{2t^2}=\frac{Iv}{R^2t}=\frac{If}{R^2}.$$

The horse-power required to overcome the inertia is H. P. $=\frac{F_1v}{550}$.

In the 290-ft. span the mass at the rack was 1231130 lbs.; suppose the time for opening be 180 seconds and the acceleration be continued over one half the time of opening. The space passed over in this time is $\frac{1}{8}2\pi R=7.6$ ft.; the time, 90 seconds; then

$$f=\frac{2\times 7.6}{90^2}=.0019$$

and

$$F_1=Mf=1231130\times .0019=2339 \text{ lbs.}$$

If the space 7.69 ft. is passed over in 90 seconds the average velocity is $7.69\div 90=.086$ ft. per second, and the maximum velocity, which is twice the average, $=.176$ ft. per second. The horse-power required is H. P. $=\frac{2339\times .176}{550}=.75$.

Case (*2*). If the acceleration extend over the time t, the

motion be then uniform for the time t_1, and the bridge is then brought to a stop in time t_2 with uniform retardation, we have

$$t = \frac{2s_1}{v}, \quad t_1 = \frac{S - (s_1 + s_2)}{v}, \quad t_2 = \frac{2s_2}{v},$$

and

$$T = \frac{S + s_1 + s_2}{v}.$$

If s_1, s_2, and either v or T be assumed the other values are easily determined:

$$f = \frac{v^2}{2s_1} \quad \text{and} \quad F_1 = \frac{Mv^2}{2s_1}.$$

It is common to assume that the power of the engine is exerted over one quarter of the 90° in producing the uniform acceleration, and that retardation takes place over the last fifth of the 90°, the motion being uniform over the remaining $\frac{55}{100}$ of the path.

It is immaterial whether linear or angular velocity and acceleration are used. The angular velocity (v_a) equals the linear velocity (v) divided by the radius of the circle at whose circumference the motion is considered as taking place: $v_a = \frac{v}{R}$; and the angular acceleration equals the linear acceleration divided by R. $f_a = \frac{f}{R}$. Under the above assumption a point at the rack-circle would pass over the distance

$$\frac{2\pi R}{4}[0.55 + (2 \times 0.25) + (2 \times 0.2)] = .725\pi R$$

in the time T if the velocity were uniform.

$$v = \frac{.725\pi R}{T}; \quad T = \frac{.725\pi R}{v}.$$

$$v_a = \text{maximum angular velocity} = \frac{v}{R} = \frac{.725\pi R}{RT} = \frac{.725\pi}{T};$$

$$T = \frac{.725\pi}{v_a}.$$

The acceleration will exist during the time

$$\frac{(2 \times .25)T}{.55 + (2 \times 25) + (2 \times 20)} = \frac{T50}{145} = .345\,T,$$

and

$$f = \frac{v}{.345\,T} = \frac{2.1\pi R}{T^2}, \quad f_a = \frac{f}{R} = \frac{2.1\pi}{T^2}.$$

In the same manner the retardation is

$$v \div \frac{40}{145}t = \frac{v}{0.276\,T}.$$

Taking the 290-ft. span again, let T be 120 seconds; then

$$v = \frac{.725\pi R}{T} = \frac{.725 \times 3.14 \times 9.8}{120} = 0.185;$$

$$f = \frac{v}{.345\,T} = \frac{0.185}{41.4} = .0044;$$

$$F_1 = Mf = 1231130 \times .0044 = 5416 \text{ lbs.};$$

$$\text{H. P.} = \frac{F_1 v}{550} = \frac{5416 \times .185}{550} = 1.8.$$

Frictional Resistances.—The following coefficients may be used for the friction of the various parts:

Sliding	friction	ϕ_s	for wedges and pivots	= 0.1
"	"	ϕ_c	" collars	= 0.07
"	"	ϕ_b	" shaft-bearings	= 0.05
Rolling	"	ϕ_v	" turntable-wheels, etc.,	= 0.003

The End-lifting Machinery and the Frictional Forces Produced by Its Operation.—The power required to lift the ends will be less than that required to turn the draw against the unbalanced wind-forces, and if a gas-engine running at high velocity be used as the motive power it will be necessary to introduce gearing sufficient to increase the time required to lift the ends to a practical amount. Considering, again, the 290-ft. span, we found that an upward force uniformly increasing from 0 at the beginning of the lift to 20000 lbs. at the end of the lift must be exerted at each wedge.

The cam arrangement shown in Plate E acts in a manner similar to the elbow-joint, which is explained on page 43 of "Plate-girder Draw-spans." Dividing the distance which the wedge moves over after it has just come to a bearing into ten parts and determining the force and its lever-arm for each of these periods, we have the results given in the following table:

TABLE L.—MOMENTS ON THE ECCENTRIC-SHAFT.

Hor. Component of Force on Wedge.	Friction of Wedge.	Force.	Force in Eccentric-rod.	Arm.	Moment.	Friction taken tangent to circum. of Eccentric. $\frac{1}{10}$ of Force in Eccentric-rod.	Arm.	Moment for Friction.	Sum of Moments.
20.000×.0×¼= .000	+20.000×.0×.3= .000=	.000	......	....	......	.. ..	...	......	......
" ×.1×"= .500	+ " ×.1× "= .600=	1.100	1.100	2.75	3.050	.110	4.69	.516	3.566
" ×.2×"=1.000	+ " ×.2× "=1.200=	2.200	2.210	2.91	6.430	.221	5.12	1.130	7.560
" ×.3×"=1.500	+ " ×.3× "=1.800=	3.300	3.320	2.97	9.860	.332	5.56	1.845	11.705
" ×.4×"=2.000	+ " ×.4× "=2.400=	4.400	4.420	2.97	13.100	.442	5.00	2.650	15.750
" ×.5×"=2.500	+ " ×.5× "=3.000=	5.500	5.530	2.91	16.100	.553	6.44	3.560	19.660
" ×.6×"=3.000	+ " ×.6× "=3.600=	6.600	6.620	2.75	18.200	.662	6.87	4.550	22.750
" ×.7×"=3.500	+ " ×.7× "=4.200=	7.700	7.720	2.50	19.350	.772	7.31	5.650	25.000
" ×.8×"=4.000	+ " ×.8× "=4.800=	8.800	8.810	2.16	19.100	.881	7.75	6.850	25.950
" ×.9×"=4.500	+ " ×.9× "=5.400=	9.900	9.900	1.56	15.450	.990	8.50	8.410	23.860

The friction acts over two surfaces of the wedge, and the amounts of the frictional forces combined with the horizontal force to drive the wedge will give the maximum force producing the moment on the driving-shaft at each period. By

the table the maximum moment on the cam-shaft is 26000 in.-lbs; it will be noticed that the friction produces more than one fourth of this moment. The moment at the engine-shaft, using a 12 H. P. gas-engine at 300 revolutions per minute, is 12 H. P. $= F 2\pi r R$, F being the force and R the number of revolutions; then $P = \dfrac{12 \times 550}{2\pi r \times 5}$ and Pr the moment at shaft $= \dfrac{12 \times 550}{2\pi \times 5} = 210$ ft.-lbs. per second or 2520 in.-lbs. per second.

Assuming that the frictional resistances will increase the power required 250 per cent, then $\dfrac{2600 \times (1 + 2.5) \times 4}{2520} =$ 140 is the number of times the power must be multiplied by the gearing to give the necessary force at the wedges. Only one half of one revolution of the wedge-shaft is required, however, to drive the wedges, and with the power multiplied one hundred and forty times we have only 42.3 revolutions of the engine-shaft to one half-revolution of the wedge-shaft. The method of determining this is as follows: The wedge-cam moves, say, $\dfrac{6 \times 3.14}{2} = 9.42$ in., and, using as small a pinion as possible (say 10 in. diameter) on the engine-shaft, the space passed over in one revolution is 31.4 in., and $\dfrac{9.42 \times 140}{31.4}$ $= 42.3$, which is the number of revolutions of the engine-shaft to one half-revolution of the wedge-cam. If the engine makes 300 revolutions per minute or 5 per second it would require only 8 seconds to drive the wedges. The working will be more satisfactory if a longer time be given the operator in which to handle the machinery, and gearing is introduced which increases the time to 13.8 seconds.

We will now find the frictional resistances of the entire line of shafting and gearing, considering one half of the bridge only.

End Horizontal Shaft (which we will call shaft No. 1).—The pressures on the shaft-bearings are:

1. The weight of the shaft, gears, etc., = 900 lbs.

2. The pressure from the eccentric-strut = 26000 × 2 ÷ 3 = 17330 lbs. (2 being the number of wedge-driving eccentrics on the shaft and 3 the radius of the eccentric).

3. The pressure from the rail-lifts. The cam which drives the strut lifting the rails has a diameter of 7 in. and the eccentricity is 1 in. The weight of the rails, etc., to be lifted is, assuming 90-lb. rails and the strut to be 6 ft. from the end of the rails, is 1120 lbs., and the struts, rods, etc., 160 lbs.; total, 1280 lbs.

4. The pressure from the gear coupling the shaft No. 1 to the short longitudinal shaft (No. 2). To determine what this pressure will be we must first find the moment on the shaft produced by the rail-lift. This will be 1280 × 1 = 1280 in.-lbs. + the friction moment of the eccentric in its collar, which is 1280 × 0.1 × 4½ = 576 in.-lbs. Then (26000 × 2) + 1280 + 576 = 53.856 in.-lbs. is the total moment, and this ÷ 14.68 (the radius of the centre gear) = 3671 lbs. as the pressure on the gear. The total pressure on the end-shaft bearings is then 900 + 17330 + 1280 + 3670 = 23180 lbs. The diameter of shaft No. 1 is 4 in., and the moment of the friction of the shaft in its bearings is 23180 × .05 × 2 = 2318 in.-lbs. This moment will increase the pressure on the gear at No. 2 shaft by 2318 ÷ 14.68 = 158 lbs., giving 3670 + 158 = 3828 lbs. as this pressure.

This gear has 2¼-in. pitch and 4-in. face. By the table page 51 of "Plate-girder Draw-spans" a tooth of this pitch will carry a load of 6320 considered as applied at the corner. Before considering shaft No. 2 we will see what the diameter of shaft No. 1 should be to safely carry its loads.

Shaft No. 1.—The total twisting moment is 26000 + (1280 ÷ 2) + (576 ÷ 2) + (2318 ÷ 2) = 28090. The bending from the gear at shaft No. 2 is 57400 in.-lbs. and from

the rail-lift 7600 in.-lbs; total, 65000 in.-lbs. By the formula on page 58 of "Plate-girder Draw-spans"

$$T' = M + \sqrt{M^2 + T^2} = 136360 \text{ in.-lbs.}$$

As the rail-lifts should be so adjusted that the lifting of the rails and the maximum work on the wedges will not be done at the same instant, the 4-in. shaft will be ample.

Gears at Junction of Shafts No. 1 and No. 2.—We found the moment transferred by shaft No. 1 to be 56200 in.-lbs., and the pressure on the teeth of the gear connecting to shaft No. 2 to be 3828 lbs. The formula for the per cent of work lost in friction by a pair of gears is

$$P_f = \pi\phi_g\left(\frac{1}{n} + \frac{1}{N}\right)Ae,$$

where $\phi_g = .15$, n and $N =$ the number of teeth in the two gears, $A =$ the length of the arc of contact, and $e =$ a constant depending on the shape and character of the teeth. For inside gears, as when the pinion works on the inner face of the rack, $\left(\frac{1}{N} - \frac{1}{n}\right)$ is to be substituted for $\left(\frac{1}{N} + \frac{1}{n}\right)$. For cycloidal teeth Ae may be taken as $.8P$ and for involute teeth as $1.2P$; $P =$ the pitch.

We have then for the pair of gears being considered

$$P_f = 3.14 \times .15(\tfrac{1}{41} + \tfrac{1}{20}).8 \times 2.25 = .065,$$

and the gear must then transmit $\dfrac{3828}{1 - .065} = 4120$ lbs.*

Short Longitudinal Shaft (shaft No. 2).—The twisting on this shaft $= 4120 \times 7.16 = 29500$ in.-lbs., and the bending will be very little, as the boxes are close to the gears. For the allowed pressure on the bearings per square inch and

* Cycloidal teeth are used above.

the arrangement that should be made to give least wear and equalize the pressures see pages 21 to 35, and also page 60 of "Plate-girder Draw-spans." This will not be discussed further here.

Gears Connecting Longitudinal Shafts No. 2 and No. 3.—The moment on shaft No. 2 divided by the radius of the gear connecting with shaft No. 3 is $29500 \div \frac{3143}{2} = 1880 =$ the tooth-pressure. The amount of work lost in friction in the shaft-bearings is $4120 + 1880 +$ (weight of shaft and gears $= 200$ lbs.) $= 6200 \times .05 \times 1.6 \div 15.71 = 31.6$ at tooth, and the per cent of work lost in the gear-friction $= .47(\frac{1}{79} + \frac{1}{27}).8 \times 1\frac{1}{4} = .023$, and $\frac{1880 + 31.6}{1 - .023} = 1960$ lbs. which is the force transferred by this gear.

Long Longitudinal Shaft (shaft No. 3).—The moment on this shaft is $1960 \times 5.23 = 10250$ in.-lbs. By the table on page 55 of "Plate-girder Draw-spans" a $2\frac{7}{16}$-in. shaft will safely carry a twisting moment of 25000 lbs. The shaft is very long, however, and the angular deflection will be the factor determining the size (see page 55) "Plate-girder Draw-spans." The formula for the angular deflection is

$$A = \frac{ML}{36000d^4} = \frac{10300 \times 145}{12 \times 36000 \times (2\frac{7}{16})^4} = .013,$$

$$360 \times .013 = 4\frac{3}{4}^{\circ},$$

which is well within the limit of deflection.

Gear at Centre of Shaft No. 3.—The diameter of this gear is 14.2 in. and the moment on the shaft 10250 in.-lbs. for each half of the bridge or the total moment $= 20500$ in.-lbs.; this divided by $7.1 = 2890$ lbs. pressure at tooth. The pressures on the shaft-bearings are then $(2 \times 1960) +$ $2900 +$ (weight of shaft and gears $= 3500$ lbs.) $= 10320$ lbs., and $10320 \times .05 \times 1.1 \div 7.1 = 80$ lbs. as force at tooth to

overcome shaft-friction. The per cent of work lost in the gears is $.47(\frac{1}{21} + \frac{1}{20}).8 \times 2\frac{1}{8} = .0799$, and $\frac{2890 + 80}{1 - .08} = 3230$ lbs. as the force to be transmitted by the gear.

By the table on strength of teeth the strain allowed on a tooth of $2\frac{1}{8}$-in. pitch is 5600 lbs.

Centre Transverse Shaft (shaft No. 4).—The moment on this shaft is $3230 \times 6.76 = 21840$ in.-lbs. Dividing this by 16.44, the radius of the gear at shaft No. 5, we have 1332 as the force at this gear-tooth. The pressures in shaft-bearings are $3230 + 1332 +$ (weight of shaft, etc., $= 400) = 4972$ lbs., and the shaft-friction $= 5000 + .05 \times 1.6 \div 16.44 = 25$ lbs. The friction at gears between shafts No. 4 and No. 5 is $.47(\frac{1}{59} + \frac{1}{19}).8 \times 1\frac{3}{4} = .046$ per cent, and $\frac{1332 + 25}{1 - .046} = 1422$ is then the force the gear transfers.

The twisting moment on the shaft is $1422 \times 16.44 = 23400$ in.-lbs., and the $3\frac{3}{16}$-in. shaft is ample.

Vertical Shaft (shaft No. 5).—The moment on this shaft is $1422 \times 5.31 = 7550$ in.-lbs.; as the shaft is long, it should not be less than $2\frac{3}{16}$ in. diameter. The force at the upper gear-tooth is $7550 \div 7.1 = 1064$ lbs. Shaft-friction $= (1422 + 1064).05 \times 1.2 \div 7.1 = 21$ lbs. There is also the collar-friction due to the weight of the shaft, etc. The common formula for collar-friction is

$$F_c = P\phi\frac{2(r^3 - r_1^3)}{3(r^2 - r_1^2)},$$

where $P =$ the pressure, $\phi = .07$, and r and r_1 are the outer and inner radii of the collar-ring. A theory based upon uniform wear of the sliding-surfaces (see page 180), and which seems the reasonable method of treating the case, gives the formula

$$F_c = P\phi\frac{(r + r_1)}{2}.$$

We have $P =$, say, 500 lbs., $\phi = .07$, $\frac{r + r_1}{2} = 2$; then $F_c = 70$ lbs. Dividing 70 by 7.1 gives 10 lbs. at the gear-tooth to overcome this friction. The per cent lost in friction at the gears connecting shafts No. 5 and No. 6 is $.47(\frac{1}{21} + \frac{1}{20}).8 \times 2\frac{1}{8} = .079$, and then $1064 + (21 + 10) \div (1 - .079) = 1200$ lbs. is the force transferred to shaft No. 6.

Shaft in Engine-house (shaft No. 6).—The moment on this shaft is $1200 \times 6.76 = 8112$ in.-lbs. The smallest shaft we will use is $2\frac{3}{16}$ in. diameter, and as there are other attachments to be made to it, it may be made larger than this. The gear at engine-shaft is 32.8 in. diameter and the pressure on its teeth is $8112 \div 16.44 = 493$ lbs. The friction of the shaft-bearings is $1200 + 493 +$ (weight of shaft, etc., $= 400$ lbs.) $= (2100 \times .05 \times 1.2) \div 16.44 = 8$ lbs. at gear-tooth. The per cent lost at the engine-gear is $.47(\frac{1}{59} + \frac{1}{19}).8 \times 1\frac{3}{4} = .046$; $\frac{500}{1 - .046} = 524$ lbs. as the force at engine-shaft gear.

Horse-power. — The engine used is a 12 H. P. gas-engine, making 300 revolutions per minute. The diameter of the engine-shaft gear being 10.62 in., the teeth pass over 33.3 in. in one revolution, and in one second $33.3 \times 5 = 166.5$ in. or 13.8 ft. per second. $\frac{524 \times 13.8}{550} = 13.1$ H. P. required. Starting with the engine-shaft, 12 H. P. $= 6600$ ft.-lbs. per second $= P \times 2\pi r \times 5.0$ (5.0 being the number of revolutions per second).

$$P = \frac{6600}{10\pi r} = \frac{6600}{10 \times 3.14 \times .44} = 446 \text{ lbs.}$$

as the allowable force at the tooth of the engine-shaft gear, against 524 as determined above. As this force is exerted only for a short interval, and not until the various parts have

acquired a certain momentum, the 12 H. P. engine does the work without undue strain.

The gearing as actually used multiplies the power nearly 640 times, and, as the weight lifted is approximately 80000 lbs., if we neglect the friction the force at the gear on engine-shaft should be $80000 \div 640 = 125$ lbs. It follows then that the force required to overcome the frictional resistance amounts to 3.2 times as much as the force required to lift the ends.

Hand Machinery for Lifting the Ends.—The force at the teeth of the gears at the point where the short vertical shaft for hand-turning connects we found to be 3230 lbs., including the friction of the gears at this point. This friction will be practically the same in the hand-turning arrangement, as the gears are the same diameter. Using a 6-ft. lever and the 13.5-in. gear, the power at the end of the lever will be $3230 \times 6.75 \div 72 = 303$ lbs. This will be increased by the friction of the shaft, which for the bearings and the collar supporting the weight of the shaft amounts to 5 lbs. The total power at end of hand-lever is then 308 lbs. Four men would be required to drive the wedges, but as the hand-power would only be required in case of breakage to the regular machinery or to the engine it is not necessary to provide for further reduction of power.

Time Required for Lifting the Ends by Hand-power.—The power is multiplied between the wedge and the hand-lever 219 times, and if the ends be lifted 2 in. the power must move over $2 \times 219 = 438$ in. $= 36.5$ ft. The men will move at the rate of, say, 3 ft. per second, and $36.5 \div 3 = 12.2$ seconds as the time required to drive the wedges.

End Wedges Driven by Rods instead of Gearing.—The arrangement for driving the end wedges shown in Plates A and B cannot be used with any power except hand-power, as the amount of motion is limited to a fraction of one revolution of

the shafts. (This is evident from the relation of the rods to the arms on the shafts.)

We will now determine how the frictional forces compare, in cases where the rods can be used, with those produced by the shafting and gearing of the arrangement just considered.

We will use a wedge having a slope of one in six and will assume that the same upward force has to be exerted (20000 lbs.). The horizontal force to produce this will then be one sixth of the vertical force. Dividing the throw of the wedge into four periods, the vertical forces at the ends of these periods are 5000, 10000, 15000, and 20000 lbs. The friction on the two sliding-surfaces of the wedge at the same periods of its movement requires horizontal forces of $P \times 0.2$, and we have for total horizontal force

At end of 1st period	$(5000 \times \frac{1}{6}) + (5000 \times .2) =$	1833 lbs.
" " " 2d "	$(10000 \times \frac{1}{6}) + (10000 \times .2) =$	3666 "
" " " 3d "	$(15000 \times \frac{1}{6}) + (15000 \times .2) =$	5500 "
" " " 4th "	$(20000 \times \frac{1}{6}) + (20000 \times .2) =$	7332 "

Laying out the toggle-joint which is used to drive the wedge in its several positions, we determine the tangential forces to be

At end of 1st period	1300 lbs.
" " " 2d "	2150 "
" " " 3d "	2000 "
" " " 4th "	0.0 "

The moment on the transverse shaft from the two wedges will then be, at the end of the second period where the tangential force is greatest, $2150 \times$ the length of the arm driving the wedge-strut $\times 2$. Making all the arms 15 in., we have $2150 \times 15 \times 2 = 64500$ lbs. as the moment which is carried by the rods to the centre of the bridge, and the strain on the rods is $2150 \times 2 = 4300$ lbs. This will be increased, however, by the moment produced by the rail-lifts and by the

friction of the rail-lift cam and of the transverse shaft in its bearings. We will consider the moment from the rail-lift and its friction the same as in the previous case, 53856 in.-lbs.

The pressures on the shaft-bearings are: from wedge-strut $2150 + 2150$ lbs., from the driving-rod arms $2150 + 2150 + (53856 \div 15) +$ (weight of rails, etc., at rail-lift = 1280 lbs.) + (weight of shaft, etc., = 900 lbs.) = 10070 lbs.

The shaft being 4 in. diameter, the moment of the friction will be $10070 \times .05 \times 2 = 1007$ in.-lbs. The moment from the rail-lift plus the friction moment from the shaft = $53856 + 1007 = 54863$ in.-lbs., which added to the moment from wedge-strut (64500 in.-lbs.) = 119363 in.-lbs., and this divided by the length of the driving-rod arms = $119363 \div 15 = 7957$ lbs. as the strain on the driving-rods. The method of proportioning the shaft for twisting and bending moments was discussed under "Plate-girder Draw-spans," page 58, and will not be considered here.

Transverse Shaft at Centre of Bridge.—The centre shaft receives the strain from the driving-rods on both arms of the bridge and the moment is then $119363 \times 2 = 239726$ in.-lbs. Rods should be used on each side of the bridge, in order to reduce the twisting moment on the shaft. All the lever-arms being the same length, the pressure on the worm-nut lever will be $7957 \times 2 = 15914$ lbs., and the pressures on the shaft-bearings $7957 + 7957 + 15914 +$ (the weight of shaft, etc., = 800 lbs.) = 32628 lbs. The moment of the friction in the shaft-bearings is $32628 \times .05 \times 2 = 3263$, and the force at worm-lever is $\dfrac{239726 + 3263}{15} = 16200$ lbs.

The adjustment should be such that the worm-lever is in a horizontal position at the point where the force required is the greatest. It will be seen that when the lever is in any other position it will produce a force having a horizontal component which must be resisted by the worm-nut and which will increase its friction as it slides in its guides.

Centre Vertical (or Worm) Shaft.—We will use a $3\frac{1}{2}$-in. worm-shaft with ball-bearings to take the end thrust, which equals the force at the worm-nut = 16200 lbs. The friction will then be rolling friction and will act with a leverage of, say, 1.8 in.; then $16200 \times .003 \times 1.8 = 87.5$ in.-lbs. is the moment of this friction, and using a 6-ft. hand-lever the force at the end of it necessary to overcome this friction moment is $87.5 \div 72 = 1.2$ lbs. By referring to page 86 of "Plate-girder Draw-spans" we see that a force of one pound at the end of a 6-ft. lever is equivalent to 161 lbs. on the nut of a $3\frac{1}{2}$-in. worm with a $\frac{3}{4}$-in. square thread (this includes the friction of the thread and of the nut sliding in the guides). Dividing 16200 by 161, we have 100.6 lbs. as the force required at the end of the hand-lever to produce a force of 16200 lbs. at the worm-nut.

The pressures in the worm-shaft bearings are, say, 105 lbs. from the hand-lever and $\frac{1}{10}$ of the vertical force at nut (this allows for the nut-lever to be somewhat out of a horizontal position). The shaft being $3\frac{1}{2}$ in. diameter, the moment of the friction in the bearings will be $\left(\frac{16200}{10} + 105\right).05 \times 2 = 172.5$, and this divided by $72 = 4.2$ lbs. at the end of hand-lever to overcome it. The total force then required at hand-lever is $1.2 + 100.6 + 4.2 = 106$ lbs.

TURNING MACHINERY AND THE FRICTIONAL RESISTANCES TO BE OVERCOME.

Wind.—An unbalanced wind-force of 5 lbs. per square foot, or, say, 60 lbs. per lineal foot for through spans and 50 lbs. per lineal foot for deck-spans, is an ample allowance for ordinary cases; the wind will be assumed to blow at right angles to plane of truss and on one arm only of the bridge.

In the 290-ft. span we have been considering the pinion and rack have a 3-in. pitch and 6-in. face; assuming the

allowed pressure on the teeth may be safely increased for the wind-forces—which will only act occasionally—to 4000 lbs. per inch of face, or 50 per cent more than the values as given in the table on page 51 of "Plate-girder Draw-spans," and assuming also that there are two teeth constantly in action, we have the strength of the gearing equal to $4000 \times 6 \times 2 = 48000$ lbs. It is useless to consider a wind-force which gives a greater stress on the pinion than this, as an attempt to turn the bridge against it would result in the wrecking of the machinery.

A pressure of 60 lbs. per lineal foot over one arm equals $60 \times 145 = 8700$ lbs. applied at the centre of gravity of the exposed surface, which is nearly at the centre of the arm; then $8700 \times 72.5 \div 9.7 = 65000$ lbs., which is the force which must be exerted at the rack to overcome the wind-pressure. This is 1.4 times as much as the teeth of the pinion and rack will withstand, even on the liberal assumption made above. If the draw is to turn against any considerable wind-force the machinery must be made more efficient, as by using two pinions. An arrangement used by Theodore Cooper, and briefly described below, is one very satisfactory method of providing suitable power. A wire rope runs around a curved guide attached to the outer end of the turntable spider-rods or axles of the turntable-wheels; one end of the rope is made fast to the curved guide and the other end is carried by means of suitable sheaves to the plunger of a hyrdaulic ram. By means of two rams motion in either direction is obtained.

Assuming that suitable machinery is provided and that the power act at a distance from the centre equal to the radius of the rack of the 290-ft. span, we will now find the H. P. required to overcome a wind-force of 60 lbs. per lineal foot acting on one arm of the 290-ft. span. Let the motion of the draw be accelerated by a force acting over one quarter of

the 90° of turning; then from page 145 we have the velocity at the beginning of the last second of the acceleration:

$$v_1 = v - f = v\left(1 - \frac{1}{.345T}\right),$$

and the space passed over in the last second of the acceleration:

$$s_1 = v - f - \tfrac{1}{2}f = v\left(1 - \frac{1}{.69T}\right).$$

The work done in this second of time is then

$$65000 \times v\left(1 - \frac{1}{.69T}\right).$$

Substituting the value of $v = \frac{.725\pi R}{T}$ (see page 145), we have the work done

$$= 65000 \times \frac{.725\pi R}{T} \times \left(1 - \frac{1}{.69T}\right).$$

If the total time of turning T be 60 seconds the work done in one second $= 65000 \times .725 \times 3.14 \times \frac{9.7}{60}\left(1 - \frac{1}{.69 \times 60}\right)$ $= 23410$ ft.-lbs. per second. $23410 \div 550 = 42.5$ H. P.

Accelerating Force (B).—We found (page 145) that if the time for turning be 120 seconds the accelerating force F_1 was 5416 lbs. If we reduce the time to 60 seconds the force will be twice this, or 10832 lbs. (B).

Friction of the Turntable-wheels (C).—The load on the turntable is, say, 70 per cent of the total weight, the remainder being carried to the centre pivot. Then 594000 $\times$ .7 $=$ 415800 lbs. is the load on the turntable-wheels, the coefficient of rolling friction is, say, .003, and 415800 $\times$ .003 $\times$ $(9 \div 9.7) =$ 1157 lbs. as the force at the rack to overcome

this friction (C). (9 and 9.7 are the radii of the drum and rack, respectively.)

Friction of the Collars on the Spider-rods (D).—Just what the force exerted by the wheels against the collars is cannot be determined. It depends upon the accuracy of the workmanship and the rigidity of the drum, etc., and perhaps most of all upon the perfect or imperfect setting of the wheel-treads. It is a common assumption to allow for a force equal to the horizontal component of the pressures on the two treads, but, as the angle is so slight that the friction could never be overcome sufficiently to allow the wheels to slide, from this cause there seems no good reason for this assumption. There is a tendency in nearly every draw, however, for the wheels to climb out of the proper circle, and some provision must be made for holding them in a truly circular path. The assumption stated above allows ample force to do this, and it will be used in determining the collar-friction. The angle of the cone is 6° 20′ and the tangent of this angle is .111. $415800 \times .111 = 46850$, which is the assumed pressure against the collars. Using the formula

$$F_c = P\phi\left(\frac{r + r_1'}{2}\right),$$

we have

$$P = 46850; \quad \phi = .07; \quad r = 2''; \quad r_1 = 4'';$$

$$F_c = 46850 \times .07 \times \tfrac{3}{12} = 820 \text{ ft.-lbs.};$$

$$820 \times (9 \div 9.7) = 761 \text{ lbs. at rack } (D).$$

Friction of Centre Pivot (E).—The load on the centre pivot is 178200 lbs. and the coefficient of friction is 0.1. The radius of the pivot is 6 in., and

$$F_P = 178200 \times 0.1 \times \tfrac{3}{12} = 4455 \text{ ft.-lbs.};$$

$$4455 \div 9.7 = 459 \text{ ft.-lbs., the force at rack.}$$

There is in addition to this the friction of the pivot in its socket; this under a heavy wind might be considerable. Any uneven tendency of the turntable-wheels to get out of position must also produce a side pressure on the centre pivot. We will assume this pressure from all causes to amount to 20000 lbs. The friction-force is then

$$20000 \times .05 \times \tfrac{6}{12} = 500 \text{ ft.-lbs.};$$

$$500 \div 9.7 = 53 \text{ lbs. at rack.}$$

The total-pivot friction is then balanced by a force of 512 lbs. at the rack (E).

Neglecting the unbalanced wind-force, we have the forces at the rack

$$B + C + D + E = 10832 + 1157 + 761 + 512 = 13262 \text{ lbs.}$$

By the table on gear-teeth a cast tooth of 3-in. pitch will carry 11238 lbs. and a cut tooth 16122 lbs. In the case of cut gears or with cast gears that have worn for a time there would probably be two teeth in action at the same time, but with new cast teeth it is more than likely that at some points of the rack all the pressure will come on one tooth.

We found that the space passed over in the last second of the acceleration was $v\left(1 - \dfrac{1}{.69T}\right) = .36$ ft. Then $13262 \times .36 = 4775$ ft. per second = the work done in the last second of the acceleration. The horse-power required $= \dfrac{4775}{550} = 8.7$.

The power is multiplied between the rack and the engine-shaft 39.77 times, and the force at the tooth of the gear on engine-shaft is then $\dfrac{13262}{39.77} = 333$ lbs. We will now see how much this is increased by the friction of the machinery between the rack and the engine-shaft.

Friction of Turning Machinery.—The per cent of work lost at the teeth of the rack and pinion is $.47(\tfrac{1}{239} + \tfrac{1}{12}).8 \times$

$3 = .099$ and $\dfrac{1 - .099}{13262} = 14735$, the force which must be exerted at the pinion-tooth. The diameter of the gear at the upper end of pinion-shaft is 37.56 in. and the force at the teeth of this gear is $14735 \times \dfrac{5.62}{18.78} = 4410$ lbs. The friction of the shaft-bearings is $(14735 + 4410) \times 2 \times .05 = 1915$. $1915 \div 18.78 = 104$ lbs., the force at tooth of 37.5-in. gear to overcome this friction. The friction on the collars from the weight of shaft and gears is $350 \times .07 \times 2.3 = 56$. $56 \div 18.78 = 2.9$ lbs. at gear-tooth. Then $4410 + 104 + 2.9 = 4517$ is the total pressure at the teeth of the gears connecting pinion-shaft and shaft to engine-house. The per cent of work lost at these gears is $.47(\frac{1}{59} + \frac{1}{13}).8 \times 2 = .07$, and $\dfrac{4517}{1 - .07} = 4858$ lbs. is the force to be exerted at this pair of gears.

The diameter of the gear at the upper end of second shaft is 32.87 in. and the force at teeth of this gear is $4858 \times \dfrac{4\,13}{16.44} = 1221$ lbs. The friction in the bearings of this shaft is $(4858 + 1221) \times .05 \times 1.62 = 492$ lbs., and $492 \div 16.44 = 30$ lbs. at gear-tooth. The collar-friction equals weight of shaft, gears, etc., $= 950$ lbs. $\times .07 \times 1.8 = 120$ in.-lbs. $120 \div 16.44 = 7.3$ lbs. at gear. Then $1221 + 30 + 7.3 = 1259$ lbs. = the force at the gears at top of second shaft. The per cent of work lost in friction at this pair of gears is $.47(\frac{1}{59} + \frac{1}{19}).8 \times 1.75 = .046$. $\dfrac{1259}{1 - .046} = 1320$ lbs. = the force required at this pair of gears.

The diameters of the gears on horizontal shaft are 10.62 and 32.87; the force at the larger gear is then $1320 \times \dfrac{5.31}{16.44} = 426.5$ lbs.

The friction of the bearings of the horizontal shaft is

$(1320 + 426 + 230).05 \times 1.6 = 158$ in.-lbs. $158 \div 16.44 = 9.6$ lbs. at the gear. Adding the friction of the gears at engine-shaft gives $\frac{426.5 + 9.6}{1 - .046} = 464$ as the force at the tooth of the engine-shaft gear.

Dividing 464 by 333, we have 1.39, or the machinery between the rack and the engine increases the power required in this case 39 per cent.

Horse-power Required.—The engine makes 300 revolutions per minute, the circumference of the gear on engine-shaft is 2.75 ft. and the force at gear-tooth is 464 lbs; then

$$\frac{300 \times 2.75 \times 464}{60 \times 550} = 11.6 \text{ H. P.}$$

Hand-turning Machinery.—As it will seldom be necessary to use the hand-power, we will assume that three minutes are allowed to turn the draw by hand.

We found on page 143 that for this time of turning and assuming that the acceleration extended over one half the motion, or for 45°, the accelerating force to be 2339 lbs. The frictional forces up to the rack and pinion we will consider the same as before, or as for turning by the engine. The unbalanced wind-force is not considered. We have for the force at the pinion-tooth $B = 2339$, $C = 1157$, $D = 761$, and $E = 512$; total, 4769 lbs. The acceleration

$$f = \frac{2s_1}{t^2} = \frac{s}{\left(\frac{T}{2}\right)^2}.$$

$s = \frac{1}{4}(2\pi R) = 15.2$ ft.; $\left(\frac{T}{2}\right)^2 = 8100$; $f = \frac{15.2}{8100} = .0019$;

v, the maximum velocity, $= \frac{2s}{T} = ft = .171$ ft. per second.

The space passed over in the last second of the acceleration is

$$v + \tfrac{1}{2}f = .171 + .0009 = .1719 \text{ ft.}$$

Friction of Hand-turning Machinery.—The per cent of work lost in friction at the rack and pinion is $.47(\frac{1}{239} + \frac{1}{12}).8 \times 3 = .097$, and $\dfrac{4769}{1 - .097} = 5280$ is the force required at pinion-tooth. The gears on the pinion-shaft are $11\frac{5}{8}$ in. and 33.8 in. diameter, and the force at the upper gear equals $\dfrac{5280 \times 5.81}{16.9} = 1815$. For the friction of the shaft-bearings we have the pressures 5280 and 1815 = 7095 lbs., and $7095 \times .05 \times 1.75 = 620.8$ in.-lbs. The collar-friction from the weight of the shaft and gears $= 550 \times .07 \times 2.2 = 84.7$ in.-lbs. $\dfrac{620.8 + 84.7}{16.9} = \dfrac{705.5}{16.9} = 41.75$ lbs. at the gear to overcome the shaft-friction. $1815 + 41.7 = 1856.7 =$ the total pressure at gear. The per cent of work lost at the second pair of gears is $.47(\frac{1}{104} + \frac{1}{47}).8 \times 1.62 = .02$, and $1856.7 \times .02 = 37.1$ lbs. $1856.7 + 37.1 = 1893.8$ lbs., the force required at gear teeth.

If a 6-ft. hand-lever be used the force at the end of it will be $\dfrac{1894 \times 12.1}{72} = 318.3$ lbs. The friction of the shaft-bearings equals $(1894 + 318) \times .05 \times 1.5 = 165.7$ in.-lbs., and the collar-friction $= 250 \times .07 \times 2 = 35$ in.-lbs. $\dfrac{165.7 + 35}{72} = 2.8$ lbs. at hand-lever to overcome this friction. The total force at hand-lever $= 318.3 + 2.8 = 321.1$ lbs.

The force is multiplied between the rack and hand-lever 17.5 times, and the space passed over by the force at rack is 15.2 ft. in 180 seconds, or .08 ft. per second, average. The power will pass over in the same time $.08 \times 17.5 = 1.460$ ft.

The maximum velocity will be twice this, or 3 ft. per second, which is the rate the men must walk during the last of the acceleration. As the men could walk faster than this for a short time, a better arrangement would have been to reduce the 24-in. pinion on the hand-lever shaft to, say, 16 in. and thus reduce the required power.

Location of Machinery.—It is very important that various gears, boxes, wheels, etc., be so placed that they are easily accessible. This is particularly true of the turntable wheels and gears. The rack and spacing-rings for the wheels are often so placed that it is next to impossible to take out a broken wheel without jacking up the entire span, which is expensive and may mean serious interruption to traffic. With proper care in the design of the treads and rack it should be possible to take out any wheel without disturbing anything but the spacing-ring. The pinion-shaft is often attached in such a manner that the replacement of a broken pinion is a slow and expensive piece of work. Attention to such points as these makes a greater saving in the operation of a draw-span than one not familiar with their working would imagine.

Material.—All important gears should be made of cast steel, the use of cast iron being limited to minor parts. Screws should be made of the best quality of forged steel, and the nuts working on them should be of some other metal, as bronze or a special grade of cast iron.

A nut and screw of the same material will not give satisfaction under high speed or heavy loads.

Pitch of Screw-threads.—The pitch of large screws should not be less than about one sixth the diameter of the screw.

Rack.—The teeth of the rack and pinions should be as large as possible and still give a good proportion to the pinion-teeth. The following table gives common proportions for the various parts of the rack.

	A	B	C	D	E	F	G
P=PITCH	NOT LESS THAN $\frac{3}{10}$p	$\frac{1}{3}$p	$2\frac{1}{2}$p	$2\frac{1}{2}$p+$\frac{1}{2}$"	$\frac{1}{4}$p	$2\frac{1}{2}$p	NOT LESS THAN $\frac{4}{10}$p

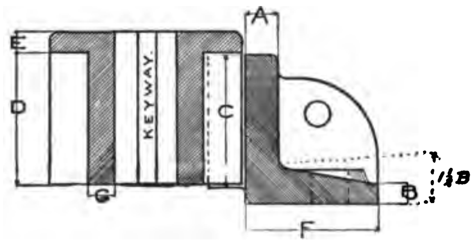


FIG. 10.

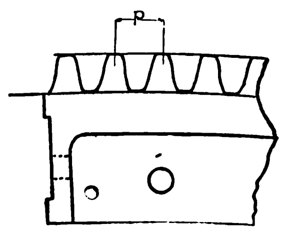


FIG. 10*a*.

Supply-tanks in Engine-house.—Whatever arrangement is used to supply the tanks in the engine-house with water must be so constructed as to be quickly and easily disconnected to allow the span to turn. Where the water-level is but a few feet below the engine-house floor a rubber tube is often used. This tube can be dropped down and taken up as required. Where the water has to be lifted to a height of 50 or 60 ft., a steam siphon or jet-pump makes a very satisfactory lift. The siphon is placed at any convenient point near the water-level. At the level of the lower deck of the bridge a coupling similar to that used on the pipe-line of an air-brake is placed, which allows the pipes to be quickly coupled or uncoupled. When uncoupled the pipes drain themselves, so there is no danger of freezing. Water may be

lifted by these pumps to a height of about 70 ft. with a steam-pressure of 50 lbs.

Coal-bins.—At some point near the boilers there should be placed a bin having a capacity of six to ten tons of coal, and having the floor built in the shape of a hopper, so that the coal will fall to one central point accessible to the shovel. Some simple hoist, driven by the engine, should be provided for filling the bins from a car on the track or from a boat alongside the pier.

Feeding the Boilers.—There should be two independent means of supplying water to the boilers: either two injectors, or, better, an injector and a pump. The tank supplying water should be placed higher than the bottom of the boiler-tubes, so that the boiler may be partly filled by gravity when under no steam-pressure.

The Engine.—An engine of ample capacity should be used. The money spent here would soon be saved in the reduced repair expense. With a double-cylinder engine of suitable capacity, if one cylinder becomes disabled the engine can still operate the bridge with one cylinder until repairs are made in the other. A single-cylinder engine of any type should not be considered in deciding upon the plant to be installed. And so far as possible all parts of the machinery of an important draw should be in duplicate.

Floor of Engine-house.—The floor in the vicinity of the boilers should be made of iron plates, or if of wood there should be an iron covering over it. A cement floor is very satisfactory.

Power for Operating Drawbridges.

Where electric current can be secured at reasonable rates, the electric motor is by far the best power for operating draw-span machinery. When not in use there is no expense for power. The amount of room required is reduced to a mini-

mum, and there is saved the expense of handling coal, ashes, etc., with the incidental advantage of freedom from dirt and smoke. The defects which rendered the earlier use of motors uncertain and unsatisfactory for this class of work have been practically eliminated, and there is probably fully as much danger of delay from breakage with the steam- or gas-engine as with the motor.

By the courtesy of the General Electric Company the results obtained with motors on several draw-spans are given below.

1. *Weight of span 525 tons, carried on rollers (rim-bearing):*

To raise one end start at 50 amp., drop to 25 amp. on 3d notch of controller.

To raise opposite end after first end is raised start at 50 amp. and raise to 65 amp. on 7th notch.

To lower end start at 60 amp. and drop to 22 amp.

To lower opposite end when first end is lowered start at 35 amp. and drop to 24 amp.

To turn draw: 1st notch, 52; 2d, 65; 3d, 90; 4th, 55; 5th, 40. With slight wind 6th and 7th notches would be used.

To lift ends, open, close, and lower ends, 4 minutes and 58 seconds are required.

2. *Weight of span 170 tons, rim-bearing, G. E. 800 motor, 6-turn armature, 520 volts.*

The trolley is suspended from radial rods.

Amperes to open, 75; on 4th notch drops to 36 amp.

Bridge has been opened and closed in 98 seconds; 2½ minutes are allowed.

Between 25 and 30 is the average number of times draw is opened daily.

Three men, eight hours each, are employed at $1100 per year per man.

3. *Weight of span 110 tons, centre-bearing and rim-bearing, G. E. 800 motor, 6-turn armature, 220 volts:*

110	amperes	to	start	1st	notch.
120	"	"	"	2d	"
100	"	"	"	3d	"
90	"	"	"	4th	"
75	"	"	"	5th	"

The trolley is suspended outside of drum, one third of circumference.

The time required to open and close, 2.2 minutes.

4. *Weight of span 120 tons, on rollers (rim-bearing):*

To open, 55 amp.

Time required to open and close, 86 seconds.

The cost of operating No. 1 is:

2 men, 12 hours each, $50 per month.....	$100.00
Light and power per month..............	52.60
Oil and repairs, cleaning, etc....	2.16
Total.........................	$154.76

The cost of current would vary in different cities: in Philadelphia it is 7½ cents per horse-power hour; in Jersey City it is 10 cents per K.W. hour, with discounts depending on amount used. This is about the same per H.P. as in Philadelphia.

Gas- and Oil-engines.—There are many locations where the use of electricity is not practicable. The majority of drawbridges are situated at a distance from electric plants. For such cases the gas- or oil-engine offers the best substitute for the motor. They are economical, can be quickly started and stopped, and as with the motor there is no expense when not in use. The same advantages of cleanliness and the absence of labor in handling fuel, ashes, etc., apply to the gas- or oil-engine as to the motor, only a small amount of

space or labor being necessary to furnish the oil or gas used. It is probable that 40 to 70 per cent of the heat supplied to the gas-engine is returned in useful work. Some of the advantages of these engines are:

1. Even the smallest engines are as efficient as the largest compound-condensing steam-engine.

2. They may be started and stopped at a moment's notice, and no energy is consumed when not running.

3. No boilers, gauges, or safety-valves are used, and hence do not require watching.

4. There are no ashes to dispose of, no chimneys are required, and the amount of skill necessary to run them is small

5. In large sizes where the gas is made on the premises, the cost is no more for the plant than for boilers. The danger of explosion is less and the operating expenses are less.

There are two classes of gas-engines: those in which the explosive mixture enters the cylinders without compression, and those in which it enters with compression. In the first class there are two types: 1st, where the explosive force is used to do work on the piston directly; and 2d, where the explosive force is used indirectly; the pressure of the atmosphere upon the piston, behind which a vacuum has been formed, being the motive force.

There are three types of oil-engines:

1. Where the vapor is converted into gas in the working-cylinder upon ignition.

2. Where the spray is converted into gas in an extension of the cylinder which may be either open to it or closed by an admission-valve.

3. Where the oil is converted into gas in a separate vaporizer heated by a lamp.

As now made most gas-engines are easily changed to allow oil to be used. The results of some tests recently made are given below.

The oil-engine is claimed by many to be even more

economical than the gas-engine. A comparison made by George Richmond is given below.

The objection that gas- and oil-engines run in one direction only is not a serious one, as simple means can be used to obtain a reversal of motion in the gearing next the engine-shaft (see Plate F).

RICHMOND GAS- AND OIL-ENGINES.

One horse-power = 2545 B. T. U. per hour.

Efficiency.	Pounds Steam.	Pounds Coal.	Cubic Feet Gas.	Pounds Oil.
100%	2.3	.3	4	.13
2%	115	15	200	6.5
4%	57.5	7.5	100	3.25
8%	28.75	3.75	50	1.625
10%	23	3	40	1.3
15%	15.5	2	26.6	.87
20%	11.5	1.5	20	.65
28%	9.6	1.25	.17	.54

	Mills.
Cost per pound of coal, at $4 per ton...........	2
Cost of gas per foot, at $1 per 1000..............	1
Cost of oil per pound, at 7 cents per gallon.....	10

BRAKE TEST OF GAS-ENGINES AT ELECTRIC-LIGHT STATION, DANBURY, CONN.

Dimensions: cylinders, 14½ in. diameter by 24 in. stroke.

	Average Number of Revolutions per Min.	Average Mean Effective Pressure.	I. H. P.	B. H. P.	Mech. Efficiency, Per Cent.	Thermal Units per Cubic Foot Gas.	Thermal Efficiency. Per I.H.P.	Thermal Efficiency. Cent. B.H.P.	Pounds of Carbon Used per I.H.P.	Pounds of Carbon Used per B.H.P.	Watt-hours per Pound of Coal.
Aver'ge of Tests.	161.9	55.65	95.29	83.3	87.31	159.03	18.61	16.25	.93	1.07	191.5

In 1894, at the annual convention of the National Electric Light Association, the committee on "data" presented a

report of fuel-consumption of twenty-four different electric-lighting stations using steam, which showed an average current output of only 91.7 watts per pound of coal. The committee for the following year presented a report of ten stations for 1895. The average taken from this table shows the average current output to be 147.5 watts per pound of coal. In large units as well as small ones the gas-engine is shown to be more efficient and economical than the steam-engine.

In one plant using coal gas the conditions were very unfavorable for good results, because the dynamos were not only inefficient, being but 81.5 per cent, but the jack-shaft consumed 35 H. P., yet the coal-consumption per electric H. P. at full load was only 2.85 lbs., while the average coal-consumption of the ten plants using steam was 4 lbs.

The results obtained from the large gas-engine at the Pantin (Paris) flour-mills show that when working at 280 indicated horse-power it consumed 1295 ozs. of producer-gas per indicated horse-power, or 1.03 lbs. per brake horse-power, which is less than 1 lb. per indicated horse-power.

The following example shows the difference in results obtained by burning gas by a jet and in a gas-engine:

"An engine using 20 cu. ft. of 16-candle-power gas per horse-power per hour is taken. This gas would furnish four 5-ft. burners of 16 candle-power each or 64 candle-power for one hour. Now this same 20 ft. of gas used in the gas-engine develops 1 horse-power per hour, and this will run ten 16-candle-power incandescent electric lights or 160 candle-power per hour. Here is a difference of 96 candle-power of light from the 20 ft. of gas in favor of the engine."

In the city of Bridgeport, Conn., the price of producer-gas is fixed at 50 cents per thousand feet, with discounts which bring the price down to one half that in the case of large consumers. At this price the cost of running a gas-

engine will vary from 1 cent to $\frac{1}{2}$ cent per horse-power per hour.—*Wilkinson.*

Now take an oil-engine of 12 indicated horse-power and let it be assumed that it is delivering 3 indicated horse-power. An oil-engine will develop an indicated horse-power on one pint of oil or about one pound in weight.

$$3 \times 24 \div 8 = 9 \text{ gallons of oil.}$$

Oil can be delivered in the mining regions, in quantities, at 15 cents per gallon. Hence the oil will cost $1.35.

We would have expended to produce 3 horse-power in a steam-engine the sum of $8.75 and for the same power in the oil-engine $1.35. In another case a steam-engine of 45 indicated horse-power, at full load, with an indicated horse-power of 7.37, using 21.25 lbs. of coal per horse-power per hour, costs $12.25 for a 24-hour run, while an oil-engine of similar power would cost $3.30. These two cases are sufficient to show the economy of the oil-engine over the steam-engine in small units.—*Prof. Unwin.*

Approximate price of gas-engines:

6 H. P.,	double	cylinder	....	$ 450
12	"	"	....	700
20	"	"	....	950
25	"	"	....	1175
30	"	"	....	1275
40	"	"	....	1400
50	"	"	....	1500

Some recent tests gave the cost of fuel for

Small steam-engines........	$\frac{2}{8}$ ct.	per	H. P.	hour.
Gas-engine.................	$\frac{1}{8}$ "	"	"	"
Oil "	$\frac{1}{2}$ "	"	"	"

The makers of gas- and oil-engines guarantee the consumption of gas per H. P. to not exceed 15 cu. ft., and for oil-engines one tenth gallon of gasoline of 74°.

These engines require little space and the danger of explosion is slight. One half-pint of gasoline is all that it is necessary to keep inside the building at one time.

Steam-engine.—There are many locations where the steam-engine will still be used. It is known to work satisfactorily and some engineers still prefer to use it. Where the engine is in nearly constant use and steam does not have to be kept up for long intervals that the engine is idle one of the greatest objections to the use of steam is removed.

The amount of steam per H. P. hour is about 25 lbs.; in the most favorable conditions it may be much less.

Makers claim to develop an indicated H. P. from 3½ to 4 lbs. of good coal, evaporating 9 to 12 lbs. of dry steam.

Steam-engines of from 5 to 40 H. P. are very costly in fuel-consumption; they are also uneconomical and inconvenient. Their low economy is clearly shown in the table below, which is the result of some experimental tests made by Prof. Unwin.

Probable I. H. P. at full load.......... ..	12	45	60	45	75	60	60
Average I. H. P. during observation.....	2.96	7.37	8.2	8.6	23.64	19.08	20.08
Coal per I. H. P. during observation.....	36.00	21.25	22.61	18.13	11.68	9.53	8.50

Tests were made in one case upon an engine which run continuously 24 hours a day. It was an engine of 12 indicated H. P. at full load, but the average was 3 H. P. on the consumption of 108 pounds of coal per hour.

$$3 \times 36 \times 24 \text{ hrs.} = 2532 \text{ pounds or } 1\tfrac{1}{4} \text{ tons nearly.}$$

Steam-coal in the mining districts (where the tests were made), delivered, is about $7 per ton; hence the cost of running this engine is $8.75 per day. It will be noticed that the amount of coal consumed per H. P. varies widely in the different tests, and to be of value all the conditions under which tests are made must be known.

The cost of operating a draw weighing about 500 tons by steam was for one month:

3 men, at $60..................	$180.00
Coal, 20 tons, at $3.............	60.00
Oil, 60 gals., at 8c..............	4.80
Total.....................	$244.80

Horse-power Required to Turn Draw, and Dimensions of Engine.

Let v = maximum velocity in feet per second at rack;
R = radius of rack;
F = force at rack;
N = the number of revolutions of the engine to one of the bridge;
n = the number of revolutions of the engine per second;
l = the length of cylinder;
a = the area of cylinder;
$2a$ = area of two cylinders;
S = speed of piston in feet per second;
P = steam-pressure (average).

Then $\text{H. P.} = \frac{Fv}{550}$.

$$Fv = 2\,Plan, \text{for one cylinder, } n = \frac{v}{2\pi R}N.$$

$$Fv = \frac{2PlavN}{2\pi R}. \quad N = \frac{2\pi RFv}{2Plav} = \frac{\pi RF}{Pla}.$$

Or, in the case of two cylinders,

$$Fv = 2PaS = 2ln. \quad n = \frac{vN}{2\pi R}.$$

$$Fv = \frac{4PlavN}{2\pi R}. \quad N = \frac{2\pi RFv}{4Plav} = \frac{\pi RF}{2Pla}.$$

If l or a be assumed the other may be found.

The mean pressure in cylinder when cut-off is made at

$\frac{1}{4}$ stroke	=	boiler-press.	$\times$ .597		$\frac{5}{8}$ stroke	=	boiler-press.	+ .919
$\frac{1}{3}$ "	=	"	$\times$ .670		$\frac{2}{3}$ "	=	"	$\times$ .937
$\frac{3}{8}$ "	=	"	$\times$ .743		$\frac{3}{4}$ "	=	"	$\times$ .966
$\frac{1}{2}$ "	=	"	$\times$ .847		$\frac{7}{8}$ "	=	"	$\times$ .992

Duty of Steam-engines.—A well-known engineer of high authority gives the following comparative figures showing the relative economy of high-grade steam-engines of different types. The results are derived from engines in actual use:

Type of Engine.	Temperature of Feed-water.	Pounds of Water Evaporated per Pound of Cumberland Coal.	Pounds of Steam per I.H.P. Used per Hour.	Pounds of Cumberland Coal Used per I.H.P. per Hour.	Cost of I.H.P. per Hour, Supposing Coal at $6 per Ton.
Non-condensing.........	210°	10.5	29	2.75	0.0073
Condensing	100°	9.4	20	2.12	0.0056
Compound jacketed.....	100°	9.4	17	1.81	0.0045

The effect of a good condenser and air-pump should be to make available about 10 lbs. more mean effective pressure with the same terminal pressure, or to give the same mean effective pressure with a correspondingly less terminal pressure. When the load on the engine requires 20 lbs. M. E. P. the condenser does half the work; at 30 lbs., one third of the work; at 40 lbs., one fourth; and so on. It is safe to assume that practically the condenser will save from one fourth to one third of the fuel, and it can be applied to any engine, cut-off or throttling, where a sufficient supply of water is available.

Cost of steam-engines (approximate):

100 H. P.	single-cylinder,	high-grade,	$1000.00
50 "	"	"	700 00
100 "	compound	non-condensing,	1750.00
50 "	"	"	850.00

A 30 H. P. double-cylinder engine and boiler complete has been installed on a bridge for $600.00.

Economy of Geared Boiler Feed-pump and Heater.—D. S. Jacobus, M.E., from experiments made by himself, calculates that the saving of fuel obtained by feeding boiler by geared pump, run from the engine, feeding water through a heater in which it is heated from 60° to 200°, is 13.2 per cent over a direct-acting pump feeding water at 60° without heater.

The saving by direct-acting pump and heater is 12.1 per cent.

The saving by injector and heater is 6.2 per cent.

The saving by injector without heater is 1.5 per cent.

There might be cases where the water required to feed the boiler might be supplied advantageously by a ram, the pipes being carried up through the centre pivot and a flexible joint used, or the pipes being so arranged as to be quickly disconnected when the draw is to be turned. The supply is practically constant and the expense almost nothing. The quantity of water which could be supplied in any special case can be determined from the rules given below.

Improved Hydraulic Rams (Rumsey).—A fall of 10 ft. from the brook or spring to the ram is abundantly sufficient to raise water to any point less than 150 ft. above the location of the machine; while the same amount of fall will also raise water to a point considerably higher, though the supply of water will be proportionately diminished as the height and distance increase. When the requisite quantity of water is forthcoming from the ram, operating under a certain fall, it is not judicious to give it more fall, for by so doing the strain on the machine is measurably augmented, those parts doing the labor are overtaxed and the durability of the apparatus impaired and lessened.

For ordinary purposes it is sufficient to say that in conveying water say 50 or 60 rods it may be safely calculated that from one tenth to one fourteenth of the water can be raised and discharged at an elevation ten times as high as the

fall, or one seventh part of the water can be raised and discharged say five times as high as the fall applied, and so in like proportion as the fall or height is varied.

Thus with a fall of 5 ft. of every 7 gallons drawn from the fountain one may be raised 25 ft., or half a gallon 50 ft. Or with 10 ft. fall one gallon of every 14 may be raised to the height of 100 ft., and so in like proportion as the fall or height is varied.

Turns in either drive- or discharge-pipe should be avoided if possible. When it is impossible to set the ram without having elbows in the pipes make the elbows as large as may be, so as to place as little obstruction to the free and easy flow of the water as is practicable. These machines are made of iron and brass. The valve and the valve-stem are made of bronze, which has more durable and lasting qualities than any other composition.

The efficiency of the hydraulic ram as a pumping-engine varies from 40 to 80 per cent of the total power of the water discharging through drive-pipe. We have found that the following rule will give a good average of quantity of water raised in a given time.

General Rule.—Multiply the quantity supplied by the spring (in gallons per minute) by 65. Multiply this product by the "head," or number of feet in fall; then divide by 100 times the height to which the water is to be elevated. The result will give the quantity of water raised per minute.

Water Required for Steam-power.—The standard, as fixed by Watt, for determining the horse-power of boilers was one cubic foot of water evaporated per hour from 212° for each horse-power. This at that time was the requirement of the best engines in use.

At the present time Prof. Thurston estimates that the water required per hour per horse-power in good engines is equal to the constant 200 divided by the square root of the pressure, and that in the best engines this constant is as low

as 150. This would give for good engines working with 64 lbs. per square inch steam-pressure an evaporation of 25 lbs. of water, and for the best engines working with 100 lbs. pressure only 15 lbs. water, per hour per horse-power.

The standard, therefore, adopted by the judges of the late Centennial Exposition at Philadelphia of 30 lbs. water per hour evaporated at 70 lbs. pressure from 100° for each horse-power is a fair one for both boilers and engines.

The table on page 179 gives the cost of operating the draw-spans on one of the most important Western railroads.

Cost of operation per month of a 185-ft. double-track railroad draw, turned an average of 7 times daily:

Coal..........................	6 tons.
Signal-oil.....................	3½ gals.
Coal-oil.......................	22 "
Machinery-oil..................	4 "
Two men, at $50................	$100.00

A 35 H. P. double-cylinder engine is used, cylinders 10 × 12 in.

Cost of operation per month of a 186-ft. double-track railroad draw, turned an average of 42 times daily:

Coal..........................	8 tons.
Signal-oil.....................	4½ gals.
Coal-oil.......................	32 "
Machinery-oil..................	8 "

2 men at $55 a month.
2 " " $45 "

A 35 H. P. double-cylinder engine is used, cylinders 10 × 12 in.

Cost of operation per month of a 250-ft. highway draw, turned an average of 55 times daily:

An 8 H. P. electric motor is used at a fixed rate of $18 per month, or $2.50 per H. P., regardless of the time in actual use. The company furnishing this power gives a rate of 7.50

TABLE M.

DESCRIPTION AND COST OF OPERATION OF DRAWBRIDGES.

Length.	Kind.	Power.	No. of Men.	Time to Open.	Time to Close.	No. of Times Opened	No. of Days Operated.	Cost of Operation. For the Year. Wages.	For the Year. Coal.	For the Year. Oil and Waste.	For the Year. Total.	Per Day Operated. Wages.	Per Day Operated. Coal.	Per Day Operated. Oil and Waste.	Per Day Operated. Total.
				min.	min.										
172′ 6″	Iron truss	Hand	4	3½	3½	2367	365	$2180.00	...	$9.00	$2189.00	$5,97	...	$0.225	$5.99
207 0	" "	Steam	2	1	1	9373	365	1860 00	$190.00	16.00	2066.00	5.10	$0.52	.044	5.67
217 8	" "	Hand	2	3½	3½	1158	365	1080.00	...	8.00	1088.00	2.96	...	.022	2.98
204 7	" "	"	1	...	...	...	273	180.00	...	...	...	.66	...	...	.66
136 4	Wood "	"	...	...	...	...	...	...	...	...	...	...	...	...	...
135 0	Iron girder	"	...	...	10...	...	...	...	...	...	...	...	...	...	...
400 0	Wood pontoon	Steam	4	6	10	378	365	2760.00	116.00	18.00	2894.00	7.56	.318	.05	7.92
405 0	" "	"	4	6	...	1167	365	3000.00	160.00	13.60	3173.60	8.22	.44	.037	8.69
191 0	" truss	Hand	1	...	...	...	215	360.00	...	...	...	1.67	...	...	1.67
115 9	Iron girder	"	1	...	...	56	215	120 00	...	...	...	.55	...	...	.55
68 0	Wood jackknife	"	1	...	...	910	240	300.00	...	...	...	1.25	...	...	1.25
123 0	Iron girder	"	1	...	...	56	215	120.00	...	...	...	.56	...	...	.56
143 0	" "	"	2	2	2	417	350	480.00	...	...	...	1.37	...	...	1.37
257 6	" truss	Steam	4	½	½	9565	365	3060.00	318.00	40.20	3418.00	8.38	.87	.109	9.36
310 0	" "	Hand	4	5	5	299	365	1224.00	...	4.20	1228.20	3.35	...	.072	3.36
359 5	" "	Steam	4	5	5	1534	365	2048.00	533.00	19.00	2600.00	5.61	1.46	.052	7.12
300 0	" "	"	3	3	3	541	365	1680.00	112.00	13.00	1705.00	4.60	.30	.036	4.94
400 0	Wood pontoon	"	3	4	7	960	365	2280.00	206.00	11.58	2497.58	6.25	.56	.031	6.84
250 0	" truss	Hand	1	...	...	...	192	320.00	...	...	...	1.66	...	...	1.66
83 6	Iron girder	"	1	...	...	...	192	100 00	...	...	...	.52	...	...	.52
248 0	Wood truss	"	2	...	...	44	192	680 00	...	...	...	3.54	...	...	3.54
142 0	" "	"	...	...	...	...	257	$5.00 per month for oiling and lighting							
201 0	" and iron	"	1	3½	3½	1583	257	600.00	...	17.29	617.29	2.33	...	.067	2.40
94 8	Iron girder	"	1	...	...	383	183	400.00	...	8.00	...	2.19	...	.043	2.83
222 8	Wood and iron	"	1	3	3	825	205	400.00	...	11.00	411.00	1 95	...	.053	2.00
68 0	Iron girder	"	...	...	...	...	...	...	...	...	...	...	...	...	...
72 0	" "	"	...	...	...	...	...	...	...	...	...	...	...	...	...
161 0	Wood truss	"	4	...	...	1904	365	2040 00	...	12 00	2052.00	5.59	...	.033	5.62
161 0	" "	"	4	...	...	3232	365	2040.00	...	12.00	2052 00	5.59	...	.033	5.62
364 0	Iron "	Steam	4	5	5	1287	365	2076.00	60 00	26 00	2162.00	5.69	.16	.071	5.92
...	30 bridges	..	60	...	...	...	...	31.928.00	...	...	...	...		...	...

cents per H. P. hour, with discounts varying on amount used and averaging about 40 per cent.

SHAFT-BEARINGS.

Pressure in Bearings.—In considering the friction of shaft-bearings there are two assumptions which may be made: first, that the pressure between the sliding-surfaces is uniformly distributed; second, that there must be uniform wear over the contact-surfaces. This second assumption would seem to be the more reasonable, and is the one made in recent investigations of the subject of journal-friction by C. G. Barth (see Proceedings of the Engineers' Club of Phila-

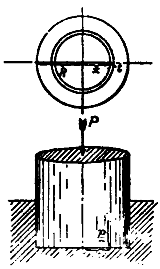


FIG. 11.

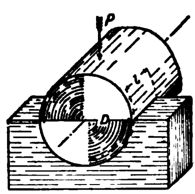


FIG. 12.

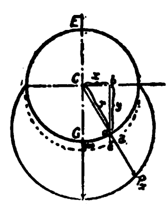


FIG. 13.

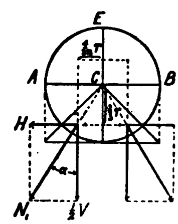


FIG. 14.

delphia for January, 1893). In an end-thrust bearing, as shown in Fig. 11, the moment of the frictional forces is on the assumption of uniform pressure

$$M_F = \tfrac{2}{3}\phi PR,$$

and on the assumption of uniform wear

$$M_F = \tfrac{1}{2}\phi PR.$$

$R =$ radius, $P =$ pressure, and $\phi =$ the coefficient of friction.

If we have an annular shaft-bearing, where R and r are

the outer and inner radii, the friction moment on the two assumptions is given by

$$M_F = \tfrac{2}{3}\phi P \frac{R^3 - r^3}{R^2 - r^2}$$

and

$$M_F = \tfrac{1}{2}\phi P(R + r).$$

Under the second assumption the maximum intensity of pressure is at the inner edge and is equal to $\frac{P}{2\pi r(R - r)}$.

Considering now the ordinary journal-bearing (Fig. 12), the intensity of normal pressure varies as the ordinate y (if we assume uniform wear), and the total normal pressure equals $N = PDl =$ a pressure uniformly distributed over the horizontal projection of the bearing of an intensity equal to the maximum intensity of pressure in the bearing. The sum of the vertical components of the normal pressures will be equal to a pressure so distributed over the horizontal projection of the bearing that the pressure at any distance x from the centre will be proportional to the ordinate y. V, the vertical component, $= \frac{\pi N}{4} = \frac{\pi p D l}{4} = P$, which is the pressure forcing the shaft into the bearing. This gives p, the maximum intensity of pressure, $= \frac{4P}{\pi D l}$. The assumption of uniform pressure would give $p = \frac{P}{Dl}$.

The sum of the horizontal components of the normal pressures, $= H$, is equivalent to a pressure so distributed over the vertical projection of the bearing that the pressure will vary as the ordinates y from the centre, being a maximum when $y = r$.

$$H = \tfrac{1}{2}prl = \frac{P}{\pi}.$$

Let N_1' = the resultant of the vertical and horizontal pressures on one half of the bearing and α the angle which it makes with the vertical; then tangent $\alpha = \frac{2H}{V} = \frac{2}{\pi}$, and $\alpha = 32° 28'$, $N_1 = .586P$.

The sum of all the moments of the friction elements is

$$M_F = \frac{4\phi Pr}{\pi},$$

whereas on the assumption of uniform pressure it is

$$M_F = \phi Pr.$$

Location of Bearings.—The life of journal-bearings may be much increased by careful and intelligent consideration of their location.

If the shaft shown in Fig. 15 transmit pressures at A and

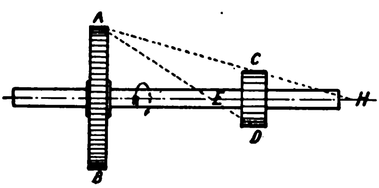


FIG. 15.

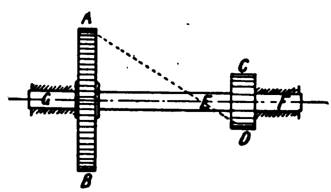


FIG. 16.

D the forces acting in the bearings will all be in the same direction and the resultant of these forces will be at E. The bearings should then either be made of equal length and placed at equal distances from E, or if placed at unequal distances their lengths should be inversely as their distance from E (see Figs. 17 and 18). If the shaft (Fig. 19) transmits pressures at A and C the reactions of the bearings will be in opposite directions and the point of application of their resultant is now at H. If it is not feasible to place the bearings on opposite sides of the centre H they should be placed as far apart as possible and one of them near H.

The bearing shown in Fig. 20 is a good one if the gears transmit at A and D, but is not as good when the transmis-

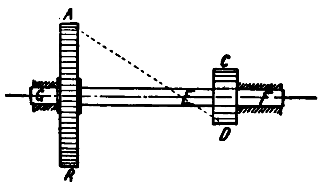


FIG. 17.

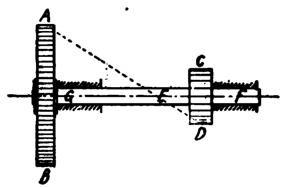


FIG. 18.

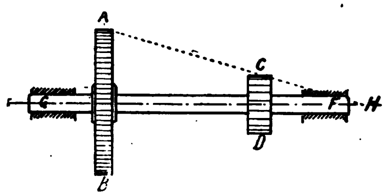


FIG. 19.

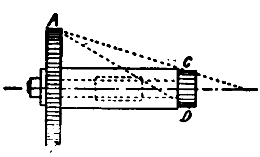


FIG. 20.

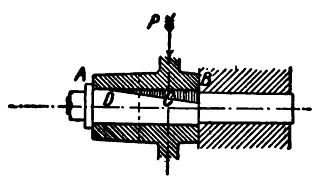


FIG. 21.

sions are at A and C. This is readily seen from the figure. When the pressures are at A and D the chamber should be nearest the gear AB.

The lengthening of a hub to one side is often done when the space is limited on the other side; if the hub be lengthened to a point D, Fig. 21, so that $DC = 2CB$, then will the pressure at B be twice what it would be if the bearing were central and of the length BD, and $1\frac{1}{8}$ times as much as it would be with a hub of length BC each side of C or with a central hub of length $2BC$. The lengthening of the hub to one side is then a real detriment instead of an advantage to the bearing.

All important bearings should be babbitted and it is often desirable to reduce the friction by means of roller- or ball-bearings.

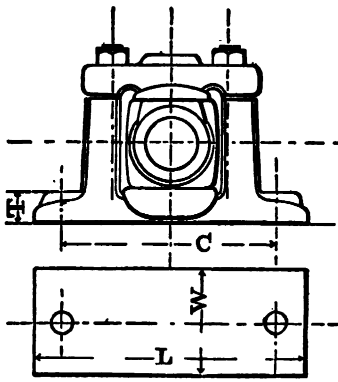


FIG. 22.

DIMENSIONS OF PILLOW-BLOCKS.*

Nominal Size of Shaft.	*L* Length of Foot.	*W* Width of Foot.	*T* Thickness of Foot.	*C* Centre to Centre of Bolt-holes.	*R* Base to Centre of Shaft.	Diameter of Bolt.
in.	in.	in.	in.	in.	in.	in.
$1\frac{1}{4}$	$6\frac{1}{8}$	$2\frac{1}{2}$	$\frac{11}{16}$	5	$1\frac{7}{8}$	$\frac{1}{2}$
$1\frac{1}{2}$	$7\frac{1}{4}$	$2\frac{3}{4}$	$\frac{3}{4}$	$5\frac{3}{4}$	2	$\frac{1}{2}$
$1\frac{5}{8}$	$7\frac{1}{2}$	3	$\frac{3}{4}$	$6\frac{1}{8}$	$2\frac{1}{8}$	$\frac{1}{2}$
$1\frac{3}{4}$	$8\frac{7}{8}$	$3\frac{5}{8}$	$\frac{15}{16}$	$7\frac{1}{8}$	$2\frac{1}{2}$	$\frac{1}{2}$
2	$10\frac{1}{4}$	$3\frac{3}{4}$	$1\frac{1}{8}$	8	$2\frac{3}{4}$	$\frac{5}{8}$
$2\frac{1}{4}$	$10\frac{7}{8}$	$4\frac{1}{4}$	$1\frac{3}{16}$	$8\frac{1}{2}$	$3\frac{1}{8}$	$\frac{3}{4}$
$2\frac{1}{2}$	$11\frac{1}{2}$	$4\frac{5}{8}$	$1\frac{5}{16}$	9	$3\frac{3}{8}$	$\frac{3}{4}$
$2\frac{3}{4}$	$12\frac{7}{8}$	5	$1\frac{7}{16}$	10	$3\frac{3}{4}$	$\frac{7}{8}$
3	$13\frac{1}{8}$	$5\frac{1}{2}$	$1\frac{9}{16}$	$10\frac{3}{4}$	4	$\frac{7}{8}$
$3\frac{1}{4}$	14	$5\frac{7}{8}$	$1\frac{5}{8}$	$11\frac{1}{4}$	$4\frac{1}{4}$	$\frac{7}{8}$
$3\frac{1}{2}$	16	$6\frac{3}{4}$	$1\frac{13}{16}$	$12\frac{1}{2}$	$4\frac{7}{8}$	1
4	18	$7\frac{1}{2}$	2	14	$5\frac{1}{2}$	1
$4\frac{1}{2}$	$20\frac{3}{4}$	$8\frac{5}{8}$	$2\frac{1}{8}$	$16\frac{3}{4}$	$6\frac{1}{8}$	$1\frac{1}{8}$
5	$21\frac{7}{8}$	$9\frac{1}{8}$	$2\frac{1}{2}$	$17\frac{1}{4}$	$6\frac{11}{16}$	$1\frac{1}{4}$
$5\frac{1}{2}$	24	10	$2\frac{5}{8}$	$19\frac{1}{2}$	$7\frac{3}{8}$	$1\frac{1}{2}$
6	$25\frac{1}{4}$	$10\frac{3}{4}$	$2\frac{7}{8}$	$20\frac{1}{2}$	8	$1\frac{1}{2}$
$6\frac{1}{2}$	$26\frac{3}{4}$	$11\frac{1}{2}$	3	$21\frac{1}{4}$	$8\frac{3}{8}$	$1\frac{1}{2}$
7	28	12	3	$22\frac{1}{2}$	$8\frac{3}{4}$	$1\frac{5}{8}$
8	32	14	$3\frac{3}{4}$	$26\frac{1}{4}$	$10\frac{5}{8}$	2

* William Sellers & Co., Standards.

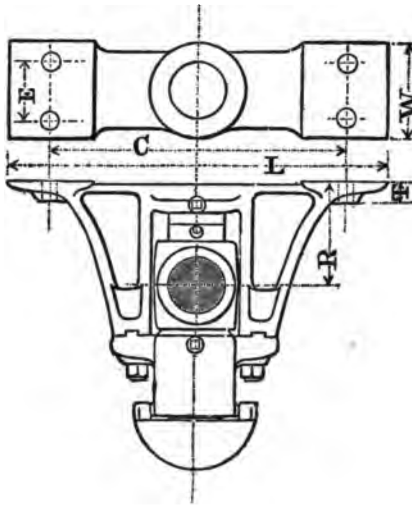


FIG. 23.

DIMENSIONS OF HEAD-SHAFT HANGERS.*

Nominal Size of Shaft.	*R* Drop.	*L* Length of Foot.	*W* Width of Foot.	*T* Thickness of Foot.	*C* Centre to Centre of Bolts.	*E* Centre to Centre of Bolts.	Number and Size of Bolts.
in.	in.	in.	in.	in.	in.	in.	in.
4	11	34	7½	1⅛	26	4¼	4—1
4½	11	35⅝	8$\frac{5}{16}$	1$\frac{11}{16}$	27⅞	5	4—1
5	11	37¼	9⅛	1⅞	28¾	5½	4—1
6	11	40½	10¼	2⅛	31½	6	4—1¼

Anti-friction Bearings.—The use of rollers or balls in some form is becoming common as a means of reducing the friction in bearings. A form especially adapted for heavy pressures and low speed is the Mossberg roller-bearing. The arrangement is very simple: a brass frame holds and separates the rollers, which are made of hardened steel and are about ⅝ in. in diameter. They run the full length of the bearing and are spaced as close as possible. If all the parts are care-

fully finished there is practically no wear and the friction is very greatly reduced.

The Hyatt Roller-bearing.—A bearing which is coming rapidly into use and which seems to have many points of merit is the Hyatt roller-bearing. It is made of a spiral coil of untempered open-hearth steel; the coils are placed as close together as practicable, but are not in contact with each other. They are made of almost any size, and are applicable to all classes of work, from the lightest to the heaviest loads, and for slow or high speed.

At the Milwaukee bridge loads of 14 tons are carried with very satisfactory results on single bearings. At Hartford a $3\frac{1}{2}$-in. shaft making 360 revolutions and carrying a 30-in. belt is supported on these bearings.

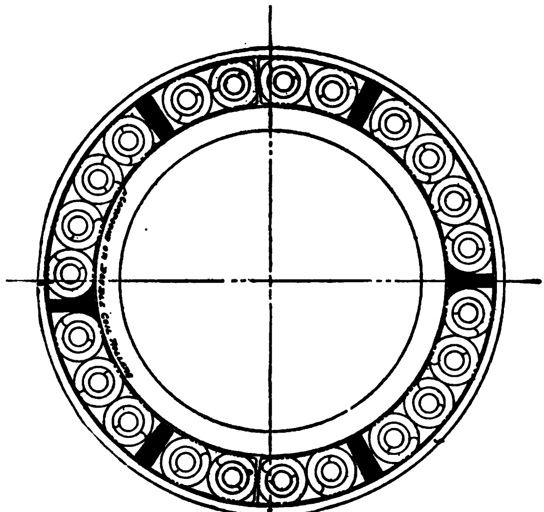
Under very heavy loads, with cast-iron or wrought-iron shaft, a sleeve of sheet steel is used over the shaft to prevent wear. The bearings are not very expensive, costing only about 50 per cent more than babbitted bearings, and they reduce the friction about 25 per cent. Compound rollers are used where the crushing force is great—that is, two coils are used, one inside the other.

The coefficients of friction as recently determined by experiments on these bearings are given below.

COEFFICIENTS OF FRICTION OF FLEXIBLE ROLLERS.

Load.	Bearing $1\frac{15}{16}''\times 3''$. Rollers $\frac{15}{16}''$.				Hyatt Steel Shafting-box. Bearing $1\frac{15}{16}''\times 7\frac{3}{4}''$. Rollers $\frac{3}{8}''$.			Bearing $8''\times 12''$. Rollers $1\frac{1}{4}''$.		
	Revolutions per Minute.									
	5	25	128	214	25	128	214	5	48	128
500 lbs.	......	......	......	......	.03156	.02367	.01972	......	......	......
1000 "	.02958	.01578	.00785	.00789	.01973	.01578	.01381	......	......	......
1500 "	......	......	......	......	.01710	.01578	.01315	......	......	......
2000 "	.01874	.00986	.00789	.00592	.01775	.01578	.01282	......	......	......
3000 "	.01249	.00789	.00789	.00526	......	......	......	......	......	......
20000 "	......	......	......	......	......	......	......	.01923	.02386	.01560
30000 "	......	......	......	......	......	......	......	.01856	.01959	.01280
40000 "	......	......	......	......	......	......	......	.01805	.01795	.01240
50000 "	......	......	......	......	......	......	......	.01708	.01692	......

FIG. 24.



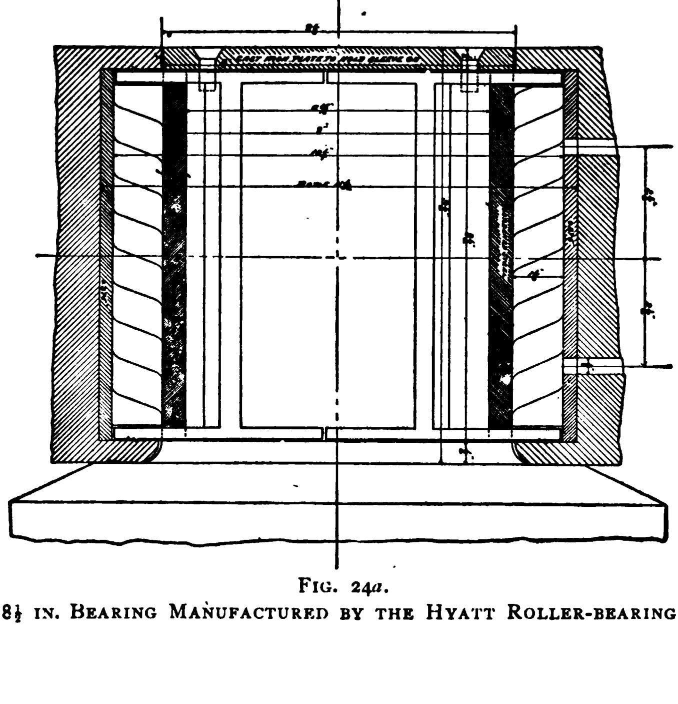


FIG. 24*a*.

8 × 8½ IN. BEARING MANUFACTURED BY THE HYATT ROLLER-BEARING CO.

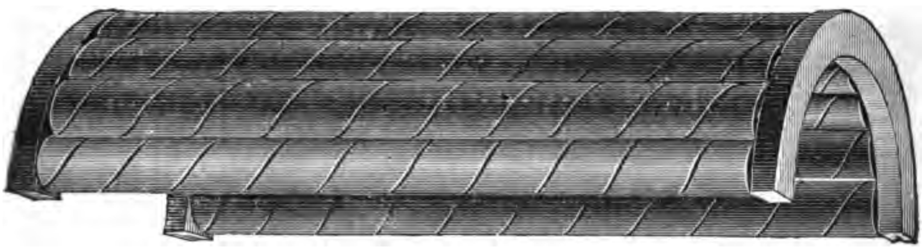


FIG. 25.
HYATT ROLLER-BEARING.

Ball-bearings.—Tests on ball-bearings have shown that the friction of these bearings is less than one third of that of the best made ordinary bearing. Under high velocity there is little heating. Balls have been made which did not vary .0001 in. in diameter. Under a load of 58330 lbs. per ball $\frac{5}{8}$-in. balls were uninjured. The bearing-surface should be very hard. A special process is used by Simonds for case-hardening; this process consists in treating steel containing little carbon in a case-hardening furnace of improved design. The steel can be case-hardened to a considerable depth, and this depth can be regulated as desired. The pressure is limited to what the bearing will stand; the balls themselves seem to be able to withstand almost any pressure uninjured.

In case shown in Fig. 26 the friction will be about 80 per cent of that in a plain bearing.

VARIOUS TYPES OF BALL-BEARINGS.

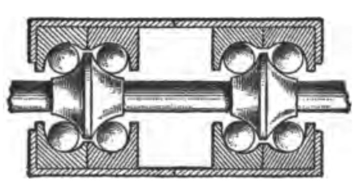


FIG. 26.
SHAFT-BEARING.

For the thrust-bearings shown in Fig. 27 the friction is 60 to 70 per cent of that in the plain bearing.

The combined bearing shown in Fig. 28 is little better than that obtained by the use of rollers alone, especially if the rollers are made of hollow spiral coils. The only advantage gained is a reduced tendency to heat.

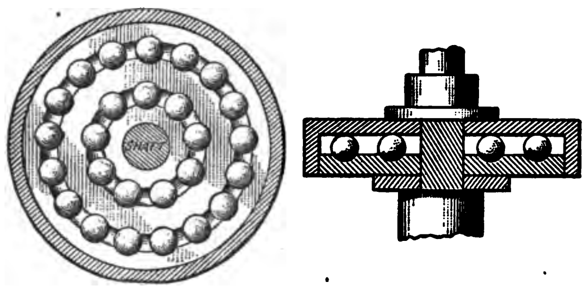


FIG. 27.
HUB THRUST-BEARINGS FOR VERTICAL SHAFTS.

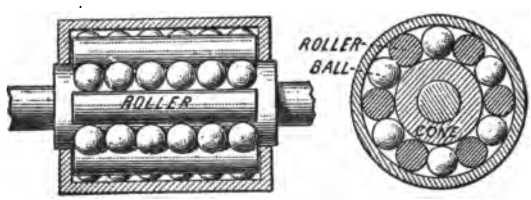


FIG. 28.
FLANGED BALL-ROLLER BEARING.

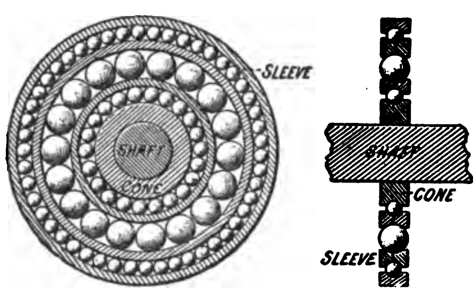


FIG. 29.
MULTIPLE SLEEVE-BEARINGS.

In the multiple ball-bearing shown in Fig. 29 the main balls bear on sleeves which in turn move on the small balls. The friction in this arrangement, it is claimed, is 8 to 10 per cent less than if the small balls were omitted.

In Fig. 30 the thrust of a worm-shaft is taken by a ball-bearing. The surfaces upon which the balls bear must be

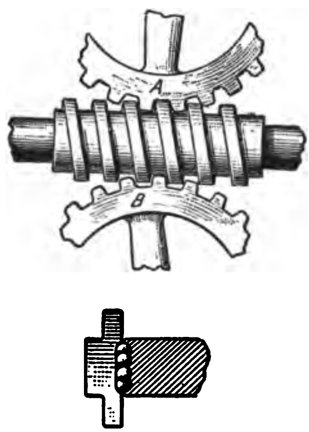


FIG. 30.
BALL-BEARING END FOR WORM-SHAFT.

made as hard as possible, or the balls soon wear grooves and the surface becomes rough and irregular.

Friction-clutch.—When the engine runs in one direction only some means of reversing the motion must be employed. The clutch shown in Fig. 31 has been used on several bridges with very satisfactory results. The clutch is manufactured by the More & White Co. of Philadelphia. It not only provides for the reversal of the motion, but it also furnishes a perfect brake, and the addition of other brakes where this clutch was used was found superfluous. By referring to Plate F it will be seen that the shaft to which the clutches attach is cut at two points, and also that the gears connecting with the engine-shaft can be thrown in and out of gear at

will; as the one set or the other is thrown into action the short section at the centre of shaft No. 6 will revolve in either direction as desired. By means of the clutches the motion is imparted to either the lifting or the turning machinery and in either direction. The action of the clutch is shown in a general way in Fig. 31. The toggle-joint furnishes a quick

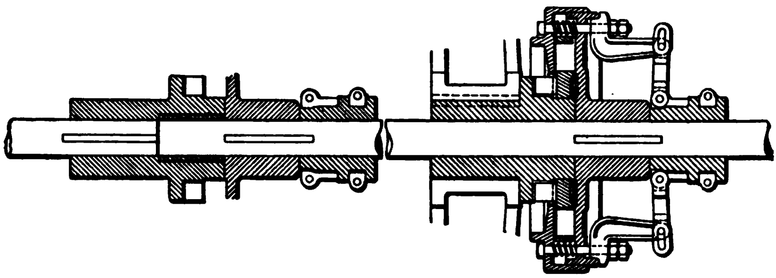


FIG. 31.
FRICTION CLUTCH.

and powerful action between the friction-surfaces. It will be seen by Plate F that two levers control the whole operation. As one set of gearing is thrown in or out of action the other set receives the opposite effect, and one lever throws either of the cut-off clutches into action or releases them both as desired. Perfect control of the operating machinery is thus secured in a simple and effective manner.

Machinery for a 397-ft. Railroad Draw (Edwin Thacher, Engr.).—Fig. 32 shows the machinery for operating both by hand and by power a 397-ft. railroad span. The upper tread for the turntable-wheels is horizontal, and, as already explained, this is a desirable feature (see page 97). The lower tread is rather thin for a casting, and to obviate the danger of breakage to some extent a wrought plate is placed between the casting and the masonry. A better plan would be to use a deeper tread, as shown in Plate E, or still better to use a tread made of bars set on edge, as shown in Fig. 35. The spacing-ring for the wheels is too light to be effective in holding the wheels to a true circle. This is a common fault and explains

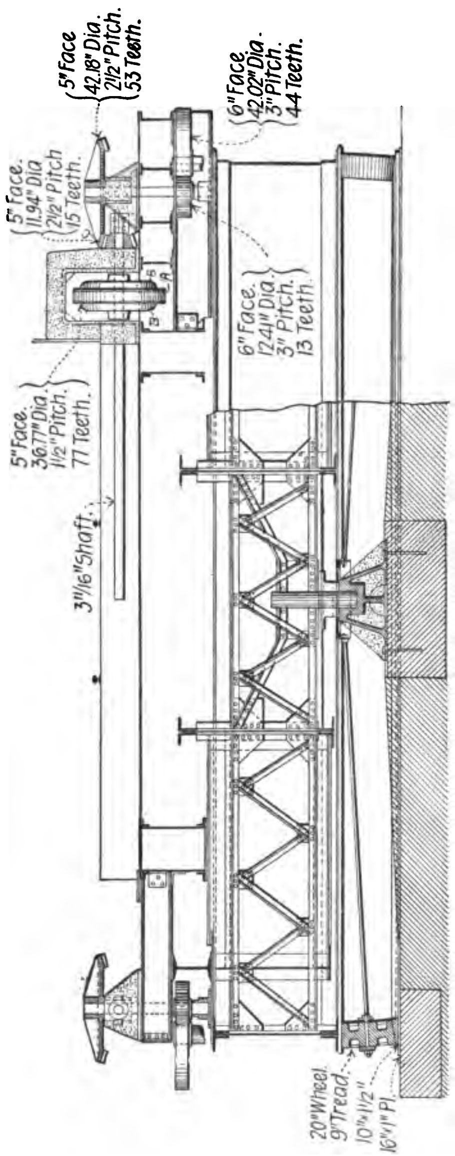

5"Face
42.18"Dia.
2½"Pitch.
53 Teeth.
5"Face.
11.94"Dia
2½"Pitch
15 Teeth.
6"Face.
42.02"Dia.
3"Pitch.
44 Teeth.
6"Face.
12.41"Dia.
3"Pitch.
13 Teeth.
5"Face.
36.77"Dia.
1½"Pitch.
77 Teeth.
3 11/16"Shaft.
20"Wheel.
9"Tread.
10"×1½"
16"×1"Pl.

FIG. 32.

the breakage of a large per cent of wheels which fail. A ring made of channels, well braced, as in Plate E, is not much more expensive, and is stiff and capable of doing its work properly. Radial rods made of gas-pipe and having a shoulder against which the wheels bear are also a much better device than a small solid rod. The hand-turning gear has provision for both slow and rapid motion, both shafts having shoulders and square ends for attaching the hand-lever. Two pinions are used in the rack for turning by power and their equal action is secured by the equalizing-gear shown in Fig. 77. Working in the gear *A* are two loose gears *B*, which revolve until the gears *C* take an equal strain. A modification of this plan is shown in Plate H, Fig. 4. This device in a slightly different form was first used probably by Shaler Smith twenty or more years ago.

The ends of the bridge are raised by independent gearing at each end of the span. Where the draw is seldom opened this is not a serious objection; but where the span has to be opened frequently the end and all other machinery should be worked from the centre. There is too much time consumed by the operator in going from one end of the bridge to the other and back again. The upright portion of the rack has very little support, and under heavy pressure the spring might be sufficient to throw all the tooth-pressure on the lower corner of the tooth, and the result would be broken teeth in rack or pinion. Brackets with a wider base placed at frequent intervals would help to give stiffness and rigidity to the rack.

Seventh Avenue Draw, New York (A. P. Boller, Engr.). Fig. 33.—This is a very heavy city bridge; there are 128 wheels under the turntable-drum, and it is evident that the most perfect workmanship and the greatest care must have been used throughout to insure the action of all the wheels. The ends are lifted by hydraulic jacks of 110 tons capacity and a motion of about 3 in. Adjustment is obtained by means of the 8-in. screws as shown. The motion of the bridge is regu-

lated by means of brakes worked by the four right- and left-hand double-threaded screws which drive the toggle attachment to the brake. Parts subject to wear are easily removed

SEVENTH AVENUE BRIDGE, NEW YORK.*

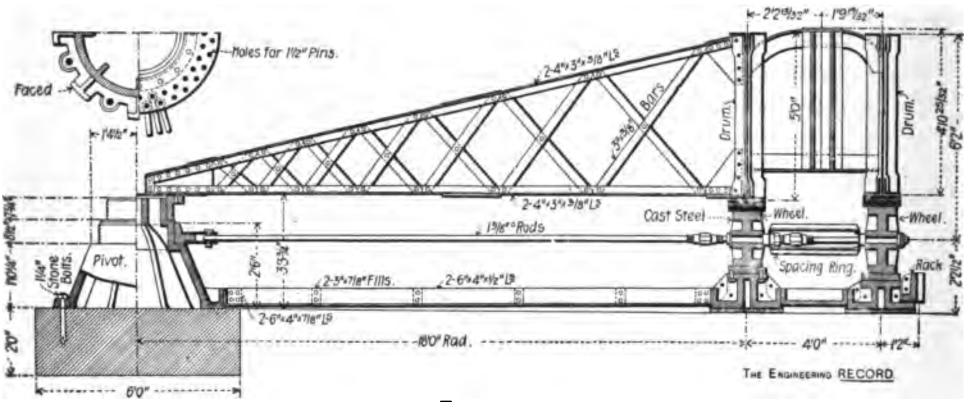


FIG. 33.
CROSS-SECTION SHOWING WHEELS, ETC.

and replaced. Mr. Boller states that as actually constructed the wheels were placed in stiff trucks, a group of four wheels to a truck. The brake device was not used, and the end-lifts were made hydraulic rams. This was the first bridge in which the double drum was ever used. The bridge works finely and has been a source of much satisfaction to its designer.

* The cuts are made from preliminary drawings, and in the actual construction some changes were made in the arrangement of the wheels and in the end machinery.

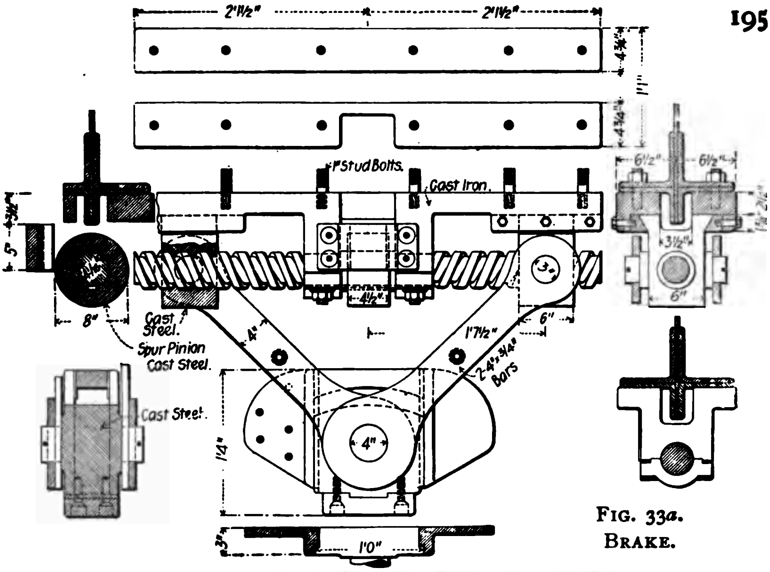


FIG. 33a.
BRAKE.

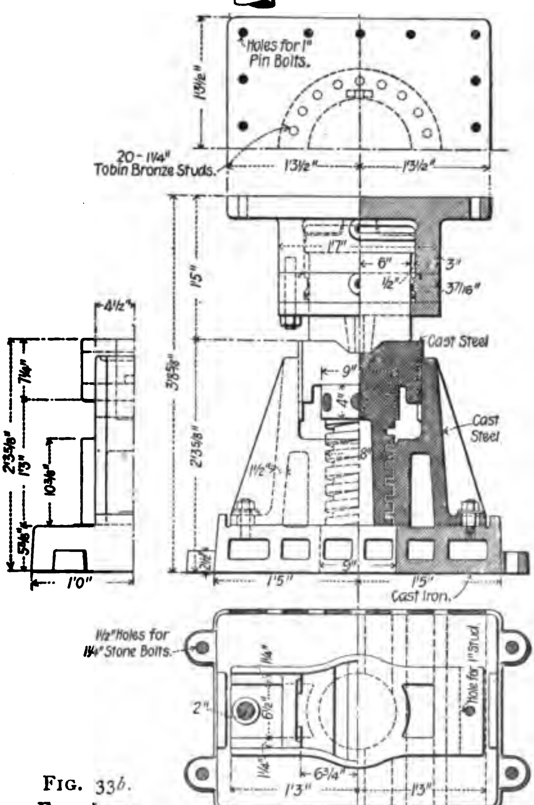


FIG. 33b.
END LIFT.

THE ENGINEERING RECORD

Machinery for a 210-ft. Railroad Draw.—Fig. 34. This design shows clearly enough that it is intended for use on an unimportant and seldom used span. Three and one half inch gas-pipe being used for shafting, the multiplication of power must be made at the ends of the bridge, and even then the most that probably could be done would be to bring the wedges to a bearing. The angular deflection would be so great and so

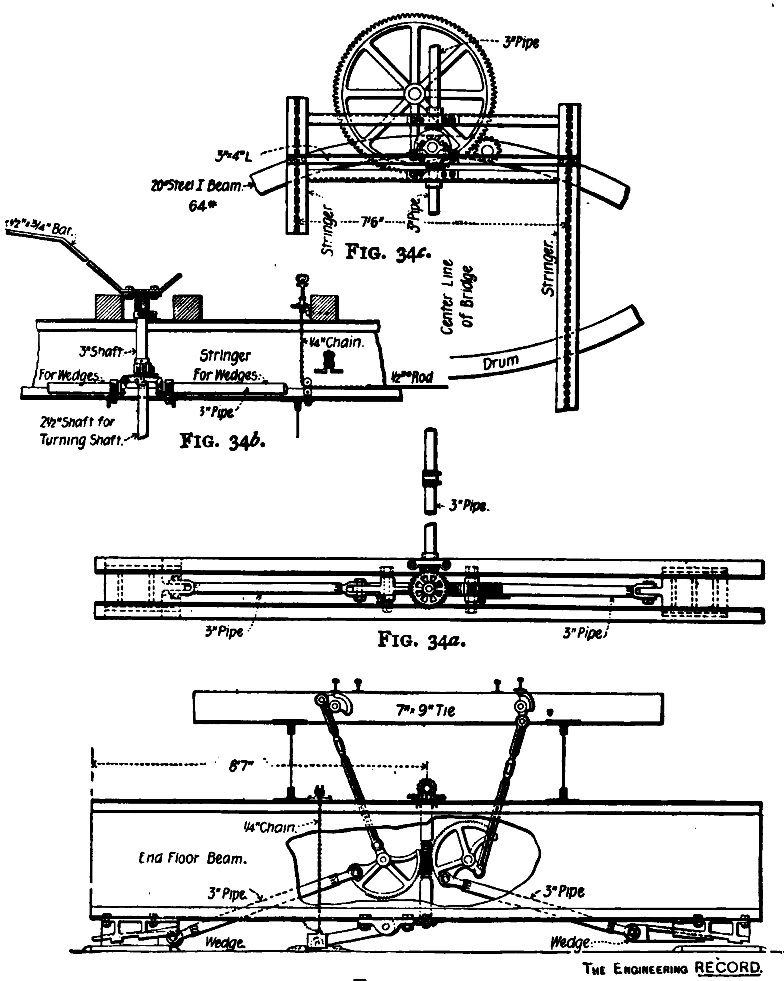


FIG. 34.

unequal that the four wedges could not be driven the same amounts after any considerable weight came on them. There was not, probably, any attempt to do more than give all wedges a bearing. The arrangement of rods and levers as shown in Plates A and B would both be much cheaper and more effective.

The support for the turning-shaft is not very satisfactory,

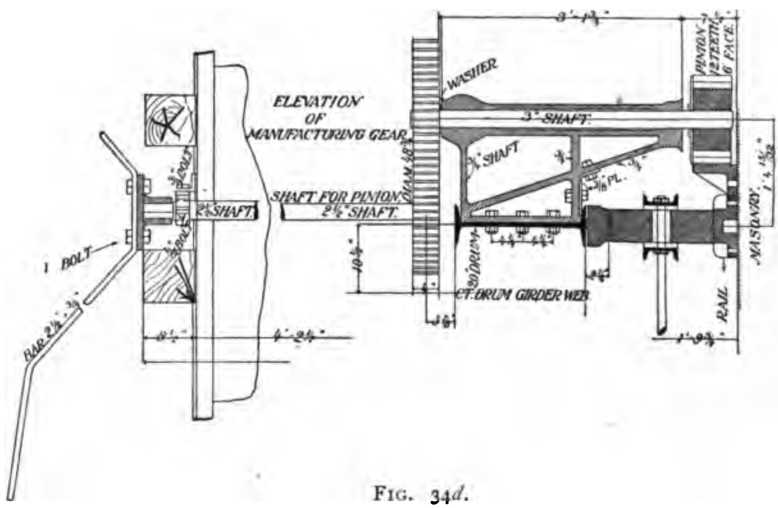


FIG. 34*d*.

as there is too great a distance unsupported above the pinion, which allows the pinion-shaft to deflect sufficiently to throw the pressure on the outer corners of the teeth. A rolled beam makes a very satisfactory drum where the load is not very great. The double-channel spacing-ring which was used is also a good feature; so also is the end-latch. Aside from the amount of clearance allowed, no dependence could be placed on this device for holding the rails in line; a separate rail-splice would be required. The latch would simply serve as a help in bringing the bridge to rest.

The friction from the rail-lift is very great; the introduction of rollers between the rail and the lifting-cam would have reduced this very much.

Machinery for a 415-ft. Railroad Draw.—Fig. 35 shows the machinery at the centre of a 415-ft. railroad draw as

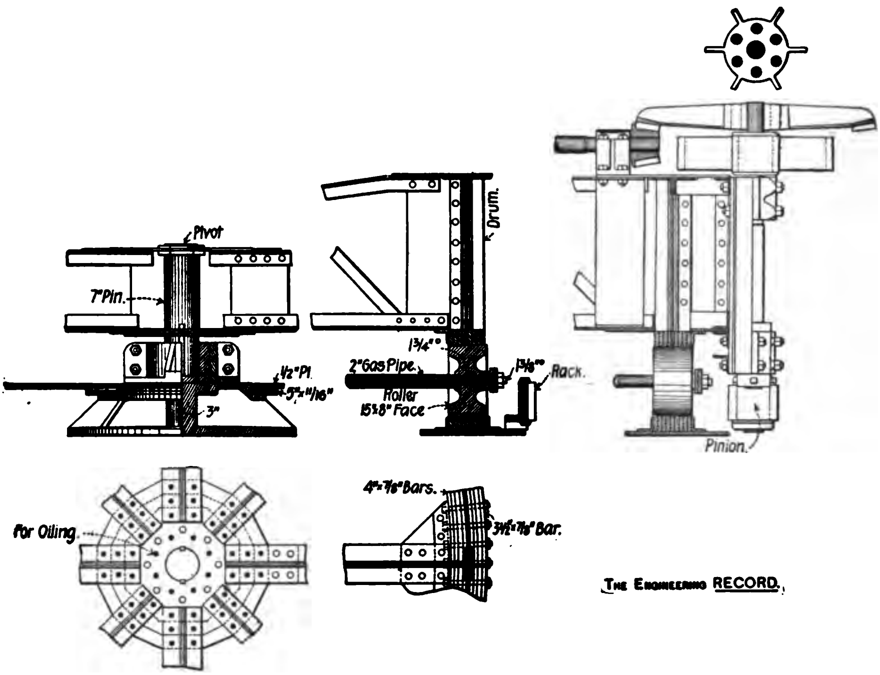


FIG. 35.

designed by the Detroit Bridge & Iron Co. The engine used was a 25 H. P. Westinghouse. The wheel-treads, the spider, and the method of distributing the load over the drum are all good features.

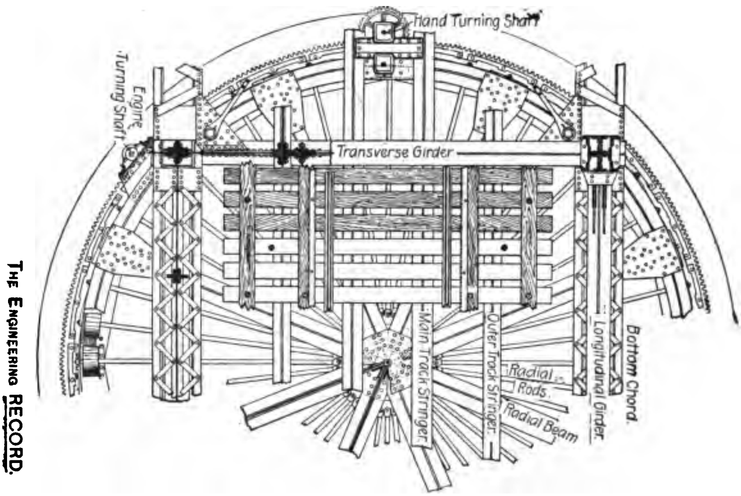


FIG. 35*a*.

Centre Bearing.—Fig. 36 shows the centre used under a heavy centre-bearing draw. The objection to all double arrangements of this kind is the uncertainty as to the equal distribution. If this is not obtained there will be broken parts and the replacement is expensive; there is also very great difficulty in getting the new parts to fit properly. Fig. 36*a* shows a simpler and cheaper method of securing an equally great bearing with much less complication. The spacing-rings used on the inner and outer ends of the rollers prevent their rubbing against each other, and in this manner the friction is reduced, as the radius of the surface where the friction now acts is less than one half of what it is without the spacing-rings.

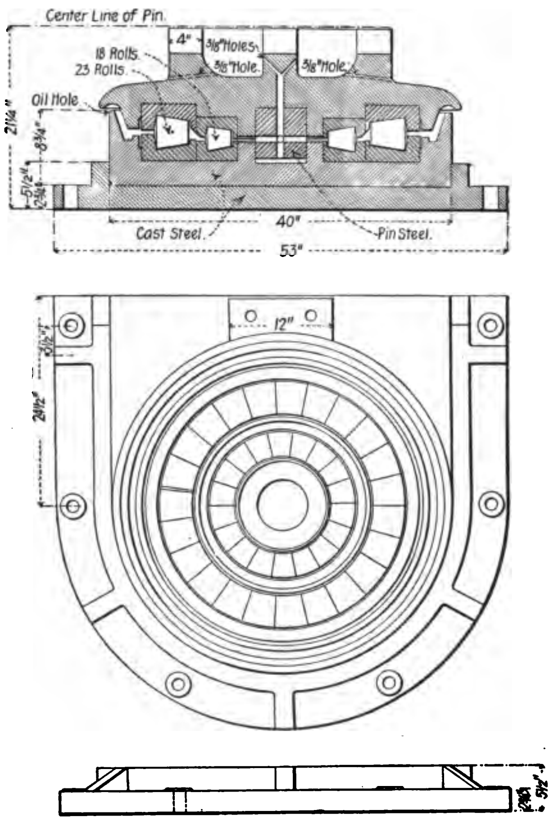


FIG. 36.

CENTRE-BEARING.

The conical roller bearing is now almost universally used on center-bearing drawbridges. It is somewhat expensive if properly made, but is reliable and efficient. It is probable that a ball-bearing will soon be made which will be more economical and as durable as the conical rollers. It may also reduce the friction somewhat.

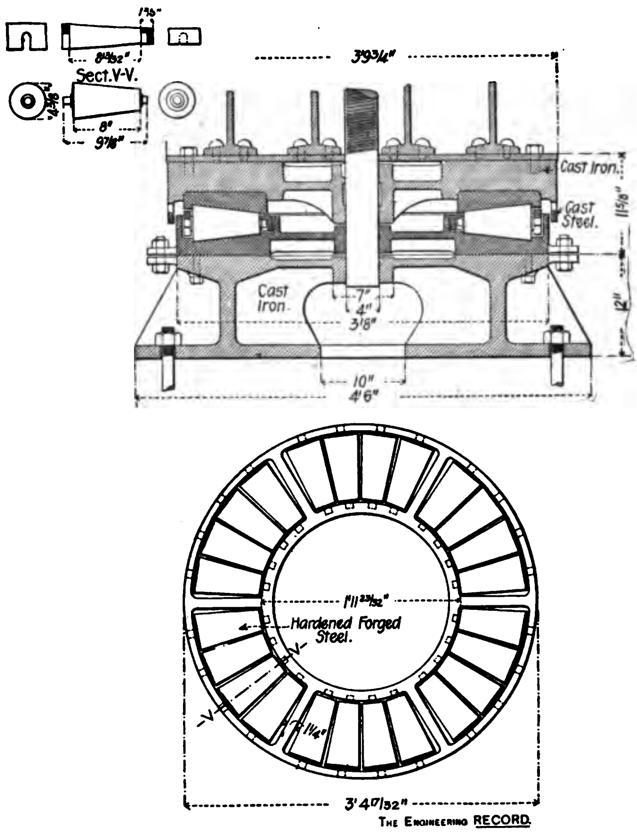


FIG. 36*a*.

CENTRE-BEARING.

Machinery for a 300-ft. Highway Draw (G. M. La Noue, Engr.).—Fig. 37 shows a design for a 300-ft. highway draw which is in most respects a very satisfactory one, the most objectionable feature being the rollers used for the end bearings. These rollers simply run up a slightly inclined plane as the span comes to rest, and the ends are but slightly raised, if at all. There is always more or less hammer at the ends

with such an arrangement, and there are so many simple and economical methods of lifting the ends that there seems little excuse for the longer use of this old-time device.

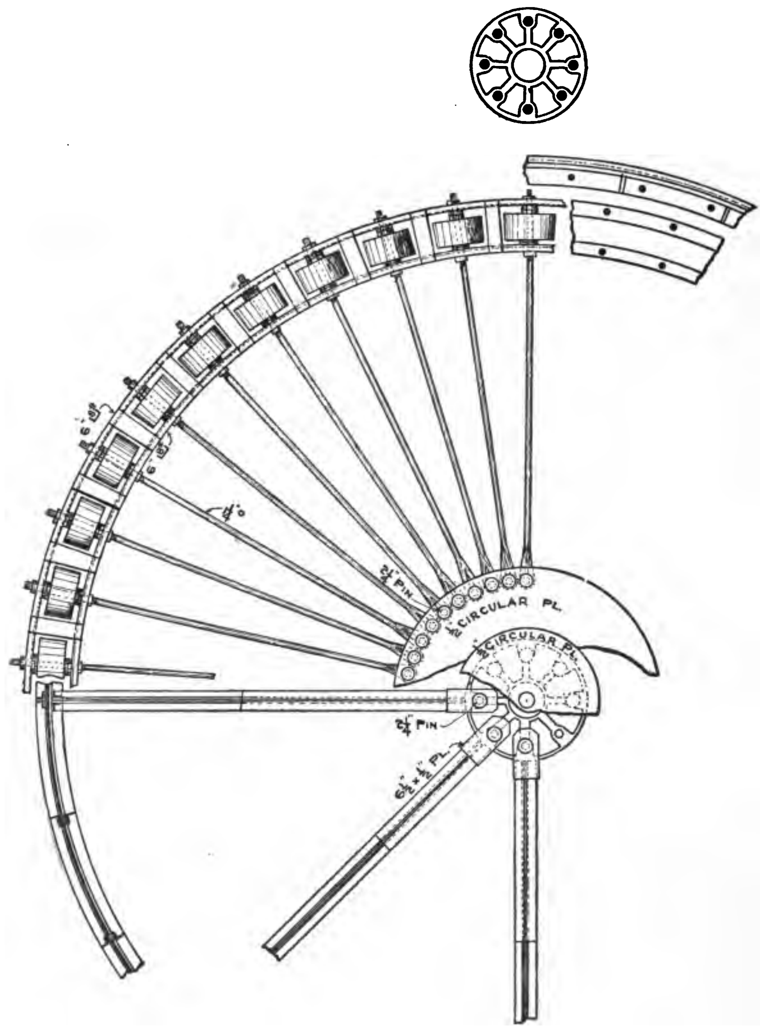


FIG. 37.

MACHINERY FOR 300-FT. HIGHWAY SPAN. G. M. La Noue. Engr.

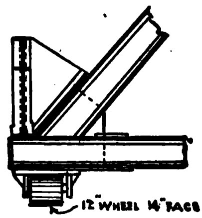


FIG. 37*a*.

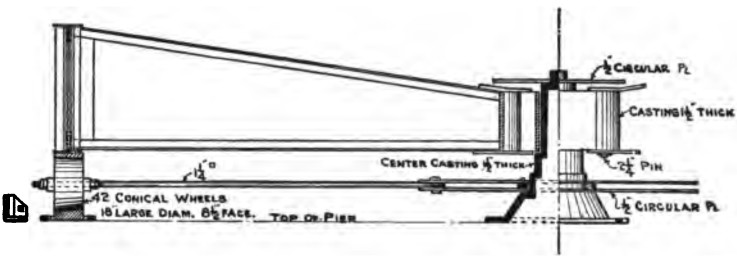


FIG. 37*b*.

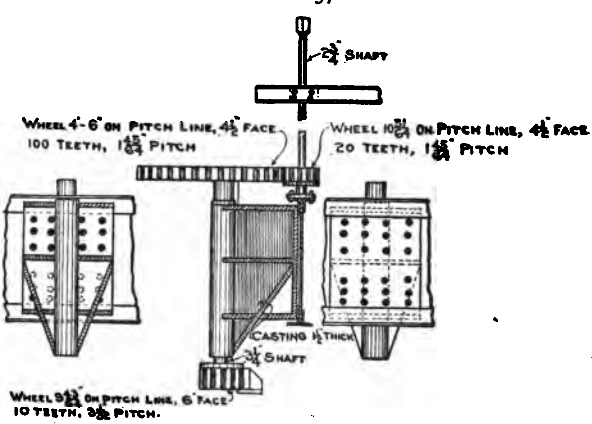


FIG. 37*c*.

End Lift, Pennsylvania Railroad Draw.—Fig. 38 shows the end-lifting machinery used by the Pennsylvania Railroad on some of its draw-spans. A similar device is also used by Theodore Cooper. The weight is not lifted by the wedge, but by the roller and toggle arrangement; as the ends are raised the wedges follow after until they reach a certain point;

beyond this point the further motion of the roller does not drive the wedge. After the roller-carriers pass the vertical the ends of the bridge drop until the wedges come to a bearing, and the rollers are free. In drawing the wedges the process is the same; the rollers lift the ends and take all weight from the wedges, and as the rollers move back the levers working the wedges are brought into action and the wedges slide out; after the wedges have moved out a suffi-

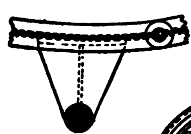


FIG. 37*d*.

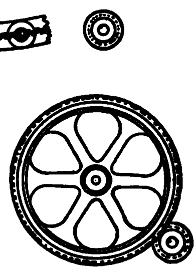


FIG. 37*e*.

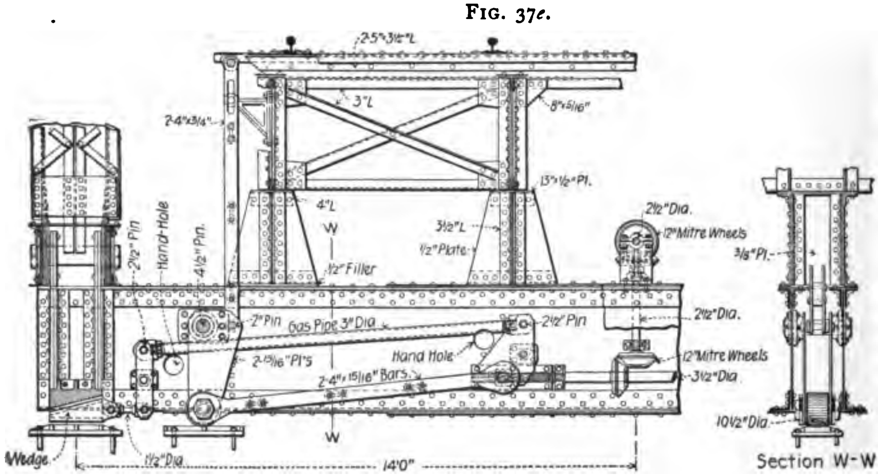


FIG. 38.
END LIFT PENNSYLVANIA RAILROAD DRAW.

cient distance the projecting lugs clear the driving-arms again and the remainder of the roller movement does not affect the

wedges. By this plan the friction is changed from sliding to rolling, and the saving in the force required to lift the ends is considerable. In the case of the 290-ft. span we found that the force necessary to overcome the friction of the wedge was about 25 per cent of the total power required.

Harlem River Draw.—Fig. 39 gives the details of a portion of the machinery of the four-track railroad draw over the

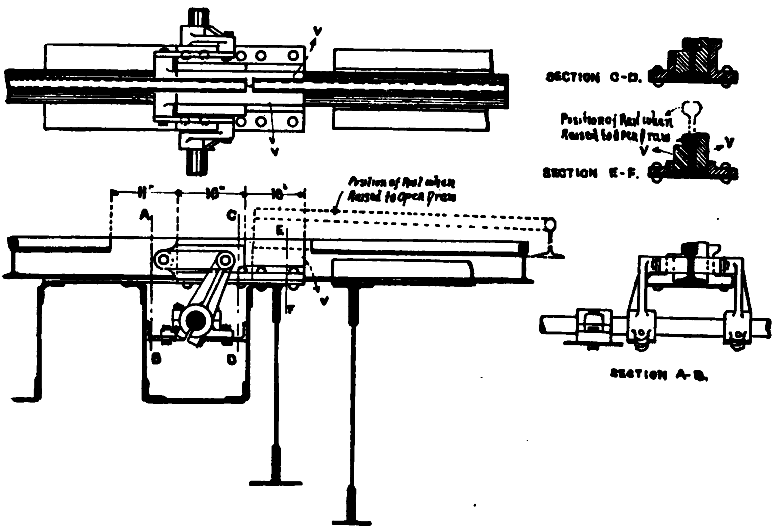


FIG. 39.

Harlem River, on the N. Y. C. & H. R. R. R. The length of the span is 389 ft. There are three trusses, spaced 26 ft. apart in the clear, carrying four tracks; a solid trough floor is

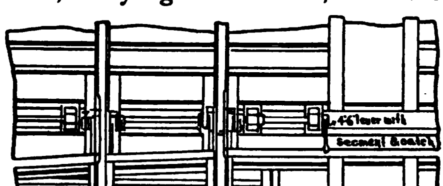


FIG. 39*a*.

DETAILS OF TRACK-LOCKING APPARATUS FOR DRAW-SPAN OF THE NEW YORK CENTRAL FOUR-TRACK DRAWBRIDGE OVER THE HARLEM RIVER.

used, with the ties resting in ballast. The weight of the

entire structure is 5000000 lbs. Two 50 H. P. engines are used for operating the turning and lifting machinery. Brake-wheels and bands control the main shaft, the bands being operated by separate steam-cylinders. Either engine will turn the bridge if necessary alone, extra gears being used in this case, which reduce the speed one half and consequently the power required. All the gearing is made of cast steel. Two minutes are required to turn the draw when both engines are used. The end-lifting machinery is designed to lift 270000 dead load and support 784000 lbs. of live load. See Fig. 78 for further details.

Brake.—As a precaution a brake of some style is usually added to the machinery of draws of considerable length. The one shown in Fig. 40 was used on the 290-ft. span shown

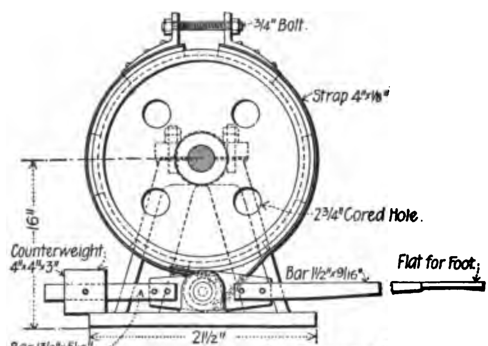


FIG. 40.

in Plates C and D; it was found, however, that perfect control of the bridge could be secured by means of the clutches used in reversing and throwing the various parts in and out of gear, so that the brake has so far proved unnecessary.

Weed Street Draw, Chicago (designed by Wm. Harman).—Two or more draws similar to the one shown in Fig. 41 have been built over the Chicago River. The floor is in two parts, hinged, as shown. When the draw is opened the two parts are lifted up and folded together in a manner similar to the closing of an open book. There is a counter-

weight at each tower, and a wire rope passes from this weight over the pulley *P* and around the cam *C*, to which it is made

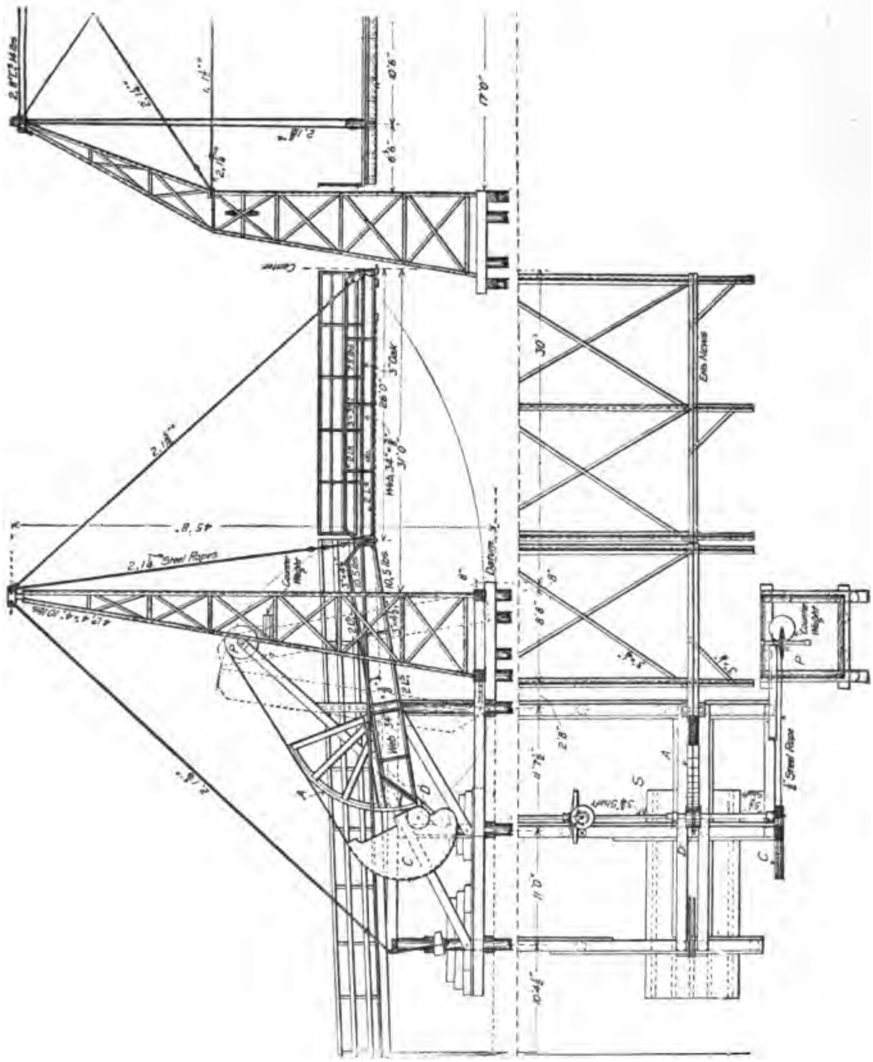


FIG. 41.

FOLDING-FLOOR DRAWBRIDGE OVER THE CHICAGO RIVER AT WEED ST., CHICAGO, ILL. Snailer & Schniglau, Engineers.

fast. On the shaft with C is a drum D; to this drum is fastened a second rope which runs over the segment A, which is attached to the rear leaf of the floor. The leverage of the counterweight is adjusted by the cam to the resistance to be overcome, so that the bridge will balance in any position. The power is applied to the shaft S by means of gearing and one man can easily handle the bridge. When open the bridge forms a perfect guard against teams, etc. The cost of operating is very small, as no engine is required and only one man to tend the bridge. The cost was about $16000, which, compared with other bridges over this river, is very low.

It would seem, however, that this type of draw is hardly suited to heavy traffic or for railroad use, as there must of necessity be a considerable amount of vibration and unequal deflection under heavy concentrated loads, particularly as there is no connection at the centre.

Folding Draw.—Fig. 42 shows the folding draw as used by the B. & M. R. R. at Boston. The girders revolve in a horizontal plane about a pivot at one end of each girder. The rails are fastened to the top of the girders by clamps, no ties being used. There are struts hinged at the ends which hold the girders parallel to each other, but do not prevent the closing together of the girders as the bridge revolves. The outer ends of the girders are held up by guy-rods, which are fastened to the top of a steel tower, the point of attachment being vertically over the axis of rotation of the girder supported, so that the girder will remain in a horizontal plane as it revolves. The girders are made of sufficient width to be rigid under the passage of the heaviest engines. The simplest method of operating is by means of cables attached near the outer ends and running to suitable drums, which are turned either by hand or by engines or motors. A rack and pinion working under a prolongation of the girders over the masonry, or a strut with a rack and pinion attachment, might be used. As no weight need be lifted, the power required is not great.

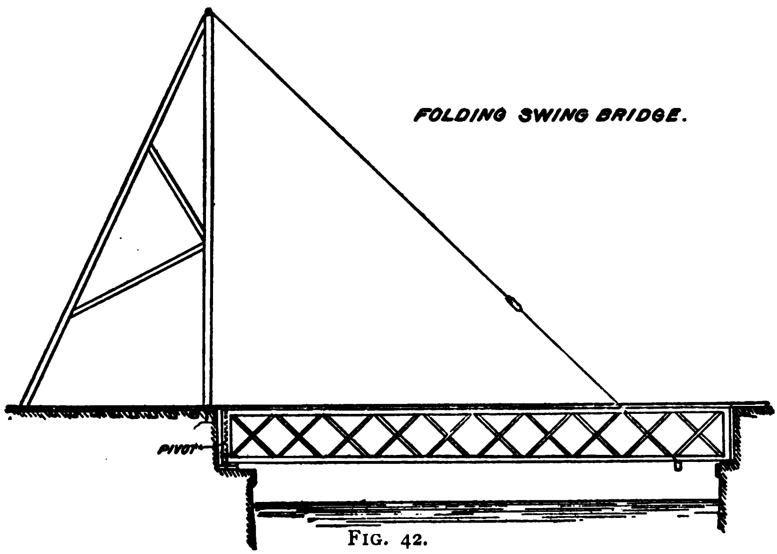


FIG. 42.

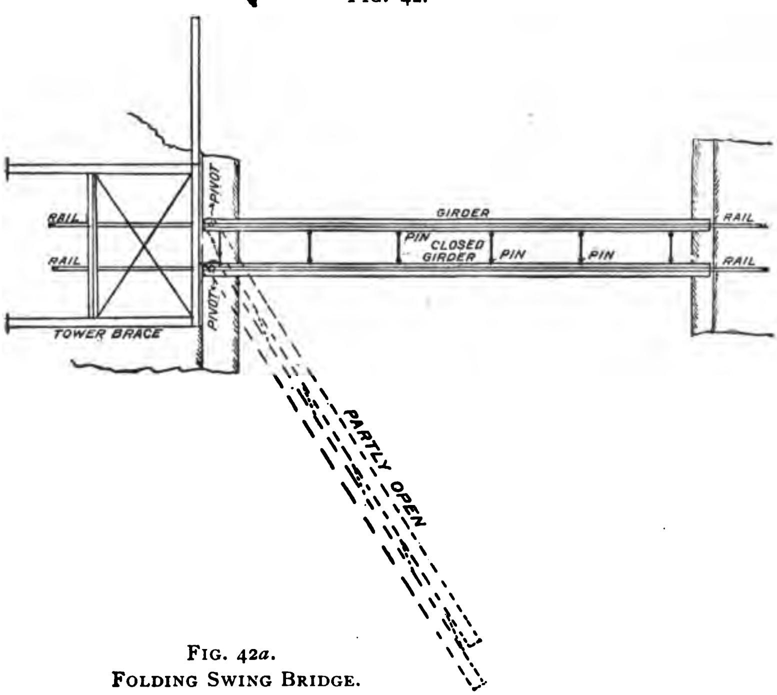


FIG. 42*a*.
FOLDING SWING BRIDGE.

If provision be made to lift the outer ends from fixed bearings before turning, and when closed to drop the ends sufficiently to take all load off the guy-rods, the power required is much increased, but a much safer bridge is secured. The rail-splice should be most carefully designed, and also the latching device made strong enough to prevent any side deflection of the ends.

Vertical-lift Bridge (J. A. Waddell, Engr.).—Fig. 43 gives a general idea of a vertical-lift bridge similar to the Halsted Street lift-bridge in Chicago, and which is fully described on page ~~58.~~ 214 The general arrangement of parts and the working of the various cables and counterweights, etc., are clearly shown.

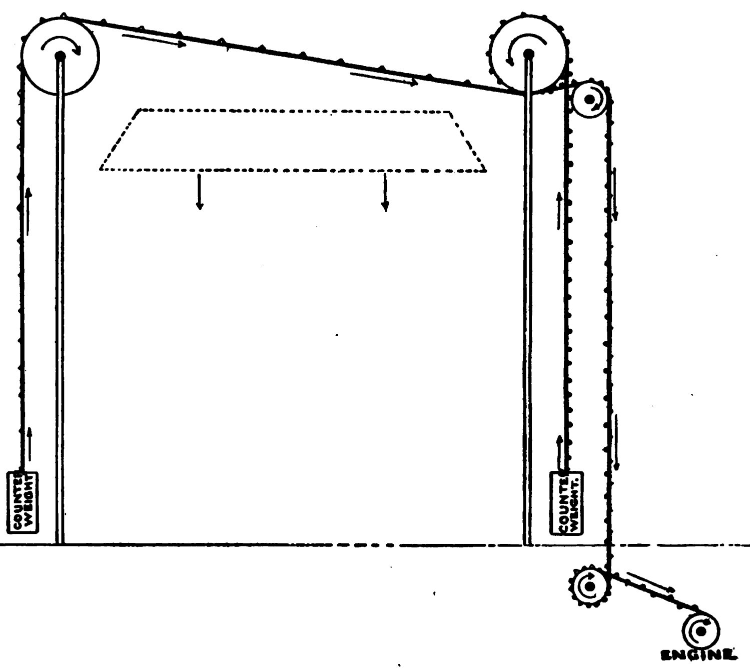
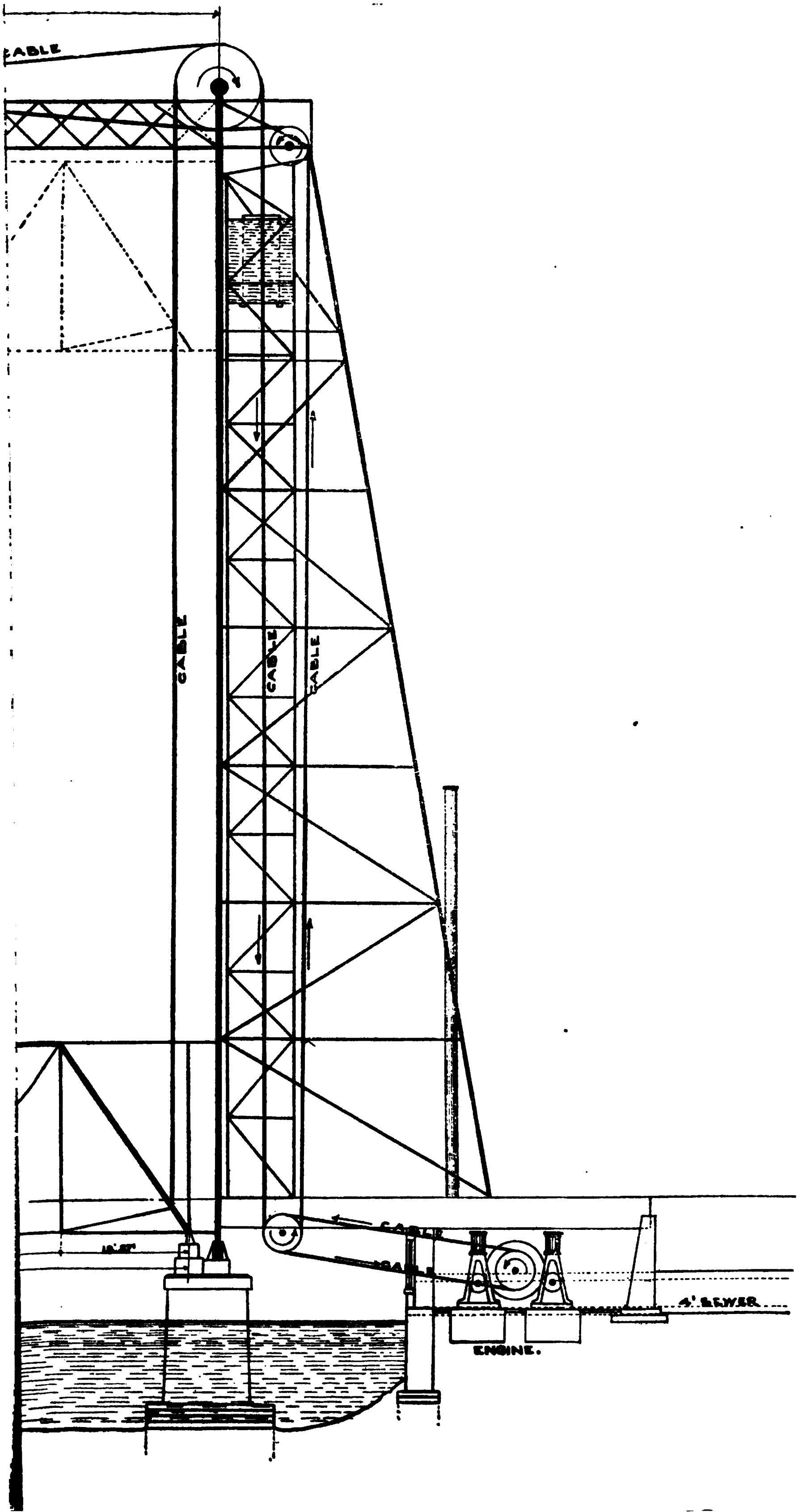


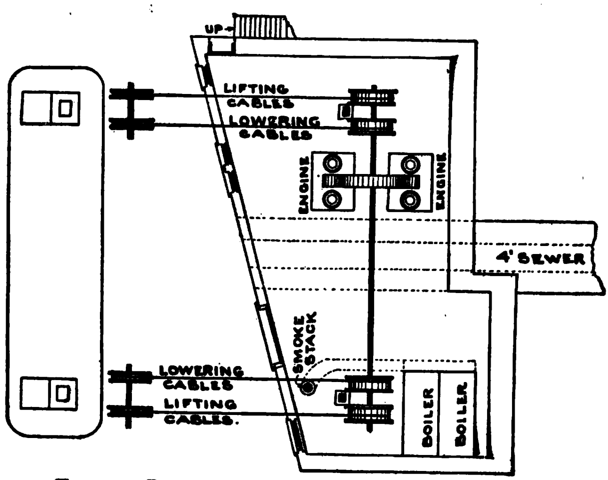
FIG. 43*b*.

DIAGRAM OF LOWERING-CABLES.



CABLE
CABLE
CABLE
CABLE
CABLE
CABLE
4' SEWER
ENGINE.

VERTICAL-LIFT BRIDGE.



PLAN OF ENGINE ROOM AND LOWER SHEAVES.

FIG. 43*a*.

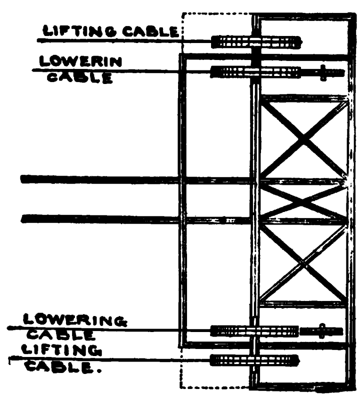


FIG. 43*d*.

PLAN OF UPPER SHEAVES.

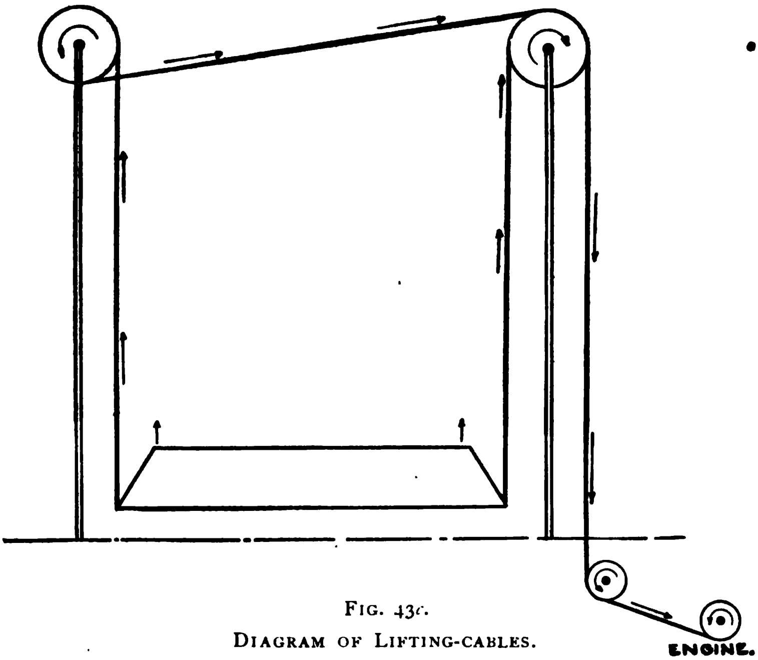


FIG. 43*c*.
DIAGRAM OF LIFTING-CABLES.

The 450-ft. Swing-span at Alton, Ill.—This bridge was designed by Geo. S. Morrison, C.E. It is a heavy double-track railroad bridge, weighing 2122000 lbs. There are 66 steel rollers 18 in. diameter and 10 in. face. If the entire weight is carried by the rollers the load per lineal inch when turning is about 3500 lbs. The engine has two 9×9 in. cylinders and runs at 126 revolutions per minute. Friction-clutches control the gearing. There are two pinions, and the force is divided between them by an equalizing-gear similar in principle to those described on page 193. The end supports are driven by a 6-in. worm with 2-in. pitch which engages a worm-gear of 16 teeth. The worm-gears are keyed to the right and left screw-shafts, which have a 1-in. thread. The worm- or screw-blocks working on these shafts slide between guides and drive the roller-struts. The rollers are double,

two at each corner of the bridge; they are 10 in. diameter and have 11 in. face; they revolve on 5-in. pins. There are two pairs of forged links 9 × 2 in. When the bridge is closed these links are vertical. The rail-lifts are operated by a bell crank and strut driven by the horizontal screw-shaft. The end-lifts serve as a lock for the bridge when closed.

The main dimensions of the boiler used are:

Shell, 60 in. diameter, 9 ft. 6 in. high.
Firebox, 48 in. diameter, 28 in. deep.
Tubes, 145, 2 in. diameter, 6 ft. long.
Heating-surface of firebox, 34 sq. ft.
Heating-surface of tubes, 329 sq. ft.
Grate-surface, 12 sq. ft.
Water-space in boiler, 77 cu. ft.
Steam-space in boiler, 23 cu. ft.

The Duluth Swing-bridge (designed by A. P. Boller).—This is a 491-ft. span, carrying two railroad tracks, two street-car tracks, and two sidewalks. The weight of the span is 1500 tons, exclusive of the wood floor, and the total mass moved is 2000 tons. There are 60 cast-steel rollers 24 in. diameter and 13 in. face. The drum is 40 ft. diameter and 4 ft. deep. The load is distributed at 16 equidistant points. Two driving-pinions are used, driven separately by a 50 H. P. motor. The motor-shaft makes 325 revolutions per minute, and the gearing reduces this 6 revolutions per minute at the pinion. The time required to turn the span is 90 seconds. Shafts and gearing are provided which enable four men to turn the bridge by hand in about 17 minutes. The maximum load on the rollers is about 50 tons each, or 7700 lbs. per lineal inch.

There is a 40 H. P. motor at each end of the bridge for driving the end-lifts, the latches, and the rail-lifts. At the centre of the cross-shaft there is a pinion driven by worm-gearing attached to the motor-shaft.

The end bearings are suspended from eccentrics and are

operated by rockers. If the motor is not stopped when the bridge is raised the pedestal rocker simply continues to lower and raise the ends until stopped; there is no danger of any of the parts being broken, as is usually the case.

The latch terminates in a roller at the bottom, and moves up and down in guides. The operation of lifting the ends raises it 4 in., and as the bridge turns it is lifted 2½ in. higher by the roller running up the inclined side of the lock-casting; when it reaches this height it is released automatically and drops 6½ in. ready to catch the lock-casting again on the return of the bridge. Powerful spiral springs placed inside the lock-castings take up any undue shock as the bridge swings in position. The cables carrying the electric current are brought up inside the centre-pivot casting and thence to the motors. All the machinery is operated from the centre of the bridge by one man.

Swing-bridge at Bowling, England.—This bridge acts as a fixed span when closed. The centre pivot has a ram of 38½ in. diameter and 3½ in. to 4 in. stroke. The span is turned by hydraulic rams of 7 in. diameter, with ball-and-socket attachments. Rollers are provided in case the ram fails to work, and in this case the turning is similar to an ordinary American bridge. There are two auxiliary rams of 28 in. diameter, equal together to the centre ram in capacity, which are used in case the centre ram is out of repair. A steam-engine of 50 H. P. is used; it has two steam-cylinders of 14 in. diameter and 16 in. stroke. The steam-pressure is 100 lbs. Working at 100 revolutions, 60 gallons of water a minute are supplied at a pressure of 900 lbs. per square inch. The centre ram will lift 486 tons. To lift the bridge 17.5 gallons of water are used and to turn the span 13.7 gallons. The time of lifting is 6 seconds and for turning 32 seconds.

The Halsted Street Lift-bridge, Chicago (designed by J. A. L. Waddell).—This bridge has a span of 130 ft., with trusses 23 ft. deep. It is designed to carry a double-track

street-railway, vehicles, and foot-passengers. The roadways are 34 ft. in the clear and the sidewalks 7 ft. in the clear. Provision is made for a vertical lift of 155 ft. above low water. When closed the clearance is 15 ft. On each side of the river is a tower 217 ft. high, at the top of which are built-up sheaves 12 ft. in diameter turning on 12-in. axles. These sheaves carry thirty-two $1\frac{1}{2}$ in. steel ropes, which sustain the weight of the span.

The counterweights are intended to just balance the weight of the span. They are composed of cast blocks strung on rods, and are held in place by guides. Wrought-iron chains attached at one end to the span and at the other end to the counterweights balance the weight of the cables, so that there is always the same weight on one side of the main sheaves as on the other. Adjustable pedestals resting in spherical seats are placed under the rear legs of the towers, and by this means the towers may, if necessary, be plumbed and the distance between the tops regulated. At the tops of the towers there are four hydraulic buffers which are capable of bringing the span to rest without jar from its greatest velocity of 4 ft. per second. Four similar buffers are placed beneath the span, one at each corner. The weight of the span complete is about 290 tons, and the cables and chains weigh about 20 tons. The total mass moved is approximately 600 tons. As the counterweights balance the span, the machinery has to overcome the friction, bend the cables, etc., and above this there has been allowed for the lifting of considerable weight if occasion should require.

Rollers placed at the top and bottom of the span steady it while in motion, and reduce the friction from wind-pressure. The expansion and contraction are taken up by heavy springs which press against the rollers and keep them tight at all times. Two 70 H. P. steam-engines are geared to an 8-in. shaft which carries the 6-ft. spiral-grooved drums over which the $\frac{7}{8}$-in. steel operating-cables pass.

The engines are provided with friction-brakes, which are always in action except when the span is being raised. Turnbuckles are used to take up the stretch in the cables. Provision is made for slowly lifting the span by hand-power if any break in the machinery should render this necessary.

The cost of operation is about $1000 per month during navigation season, which is very high. There are three engineers, two signal-men, four policemen, and an oiler. This number could be much reduced by the use of electric motors. The opinion of many engineers is that the cost of maintenance will be heavy, the renewal of 14000 ft. of wire rope being one important item.

The ability of the bridge to withstand collisions would seem to be much better than that of the bascule type.

There has been considerable doubt as to the efficiency of the ball-and-socket adjustment used on the rear columns of the towers, and there would seem to be other surer and less expensive means of accomplishing this result.

Allowed Stresses.

The load on the 1½-in. cables is 18750 lbs.

Pressure on the journal-bearings, 600 lbs. per square inch.

Pressure on the buffers, 200 lbs. per square inch.

Shafting, combined twisting and bending, 10000 lbs. per square inch.

Assumed dead load, 4100 lbs. per lineal foot for the trusses.

Assumed live load, 4500 lbs. per lineal foot for the trusses.

Assumed live load, 100 lbs. per square foot for the floor.

Assumed live load, 11 tons on single 6-ft. roller for the stringers.

Maximum velocity of span, 4 ft. per second.

Some of the calculations made by Mr. Waddell are as follows:

The Buffers.

Moving weight, 1200000 lbs.

Moving mass, $1200000 \div 32.2 = 37267$.

Maximum velocity, 4 ft. per second.

Energy due to same, $37267 \times (4)^2 \div 2 = 298136$; say 300000.

Number of buffers, 4.

Energy per buffer due to moving mass, $300000 \div 4 = 75000$ ft.-lbs.

Stroke of buffer, 4 ft.

Constant pressure on piston, $75000 \div 4 = 18500$ lbs.

Diameter of cylinder, 12 in.

Area of cylinder, 113 sq. in.

Area of four holes, 4 sq. in. nearly.

Net area of piston, $113 - 4 = 109$ sq. in. $= A$.

Intensity of pressure on piston $= 18750 \div 109 = 172$ lbs.

Hydraulic head due to 172 lbs. pressure $= 396$ ft.; say 400 ft.

Formula for velocity through holes:

$$v' = 0.7 \sqrt{2gh} = 0.7 \times 8 \sqrt{400} = 112 \text{ ft. per second.}$$

$v =$ velocity of piston at any part of the stroke; its value will diminish uniformly from 4 ft. per second to zero. $A' =$ net area of the four orifices for the position of piston corresponding to the varying velocity v. Then

$$Av = A'v' \quad \text{and} \quad A' = \frac{Av}{v'} = \frac{109}{112}v = 0.973v.$$

For $r = 4$, $A = 3.89$ sq. in., and $\frac{A'}{4} = 0.973$ sq. in., and in similar manner values of $\frac{A'}{4}$ are found for $v = 3, 2, 1$, and 0.

Power.

The amount of power required is dependent upon the coefficient of friction in the journals of the main sheaves. The value of this coefficient was assumed as .05.

The three following cases were investigated:

1. No wind acting, balanced loads, and a maximum velocity of 3 ft. per second.
2. No wind acting, balanced loads, and a maximum velocity of 4 ft. per second.
3. Greatest assumed wind-pressure and a maximum velocity of 2 ft. per second.

This velocity will, it is assumed, lift the span 85 ft. in 50 seconds.

Case No. 1.—Load on journals, 1320000 lbs.

Frictional resistance of journals, $1300000 \times .05 = 66000$ lbs.

Velocity of axle in journal, 0.25 ft. per second.

Work of friction, $66000 \times 0.25 = 16500$ ft.-lbs.

Horse-power, $16500 \div 550 = 30$.

Inertia: assume that in 15 ft. the full velocity of 3 ft. per second will be developed.

$$\text{Mass} = \frac{1200000}{32.2} = 37267.$$

$$\text{Kinetic energy} = \frac{37267}{2} \times (3)^2 = 167700 \text{ ft.-lbs.}$$

The average velocity is 1.5 ft. per second. The time required for development, 10 seconds.

$$\text{Energy expended per second} = \frac{167700}{10} = 16770.$$

$$\text{Corresponding H. P.} = \frac{16770}{550} = 30.5.$$

For bending the cables at a velocity of 3 ft. per second 6 H. P. approximately are required.

The total H. P. is then $30 + 30.5 + 6 = 66.5$ H. P.

Case No. 2.—The work of friction will be proportional to the velocity; then in this case the H. P. for friction will be $30 \times \frac{4}{3} = 40$.

Inertia: energy developed $= \dfrac{37267 \times (4)^2}{2} = 298136$ ft.-lbs.

The average velocity during development = 2 ft. per second, and the time = 10 seconds.

Energy per second $= \dfrac{298136}{10} = 29813$.

H. P. $= \dfrac{29813}{550} = 54.3$.

For bending the cables at velocity of 4 ft. per second, approximately 8 H. P.

The total H. P. $= 40 + 54.3 + 8 = 102.3$.

To attain maximum velocity, 10 seconds.

Duration of maximum velocity, 29 seconds.

To overcome maximum velocity, 2 seconds.

Total time, 41 seconds.

Case No. 3.—Friction: H. P. as in Case 2. $\dfrac{40}{2} = 20$.

Inertia: the energy developed $= \dfrac{298136}{4} = 74534$ ft.-lbs., and the time 10 seconds. $\dfrac{74534}{10} = 7453$ ft.-lbs. $= 13.6$ H. P.

For bending the cables at a velocity of 2 ft. per second, 4 H. P. approximately.

Unbalanced load: energy $2000 \times 2 = 4000$ ft.-lbs., and the corresponding H. P. $= 7.3$.

Wind-pressure: total wind-pressure on span = 50000 lbs.

Diameter of rollers = 15 in.

Diameter of axle = 5 in.

Velocity of axle $= \frac{5}{15} \times 2 = 0.67$ ft. per second.

Coefficient of friction, 0.05.

Frictional resistance $= 50000 \times .05 = 2500$ lbs.

Work of friction $= 2500 \times 0.67 = 1675$ ft.-lbs.

Corresponding H. P. $= \frac{1675}{550} = 3.$

To this should be added the rolling friction = I. H. P. for the rollers, making total friction for the rollers 4 H. P.

Wind-pressure on the counterweights: area exposed, $4 \times 8 \times 10 = 320$ sq. ft.

Pressure on this area, $320 \times 30 = 9600$; say 10000 lbs.

Coefficient of friction $= 0.15$.

Frictional resistance $= 10000 \times 0.15 = 1500$ lbs.

Work of friction $= 1500 \times 2 = 3000$ ft.-lbs. per second.

Corresponding H. P. $= \frac{3000}{550} = 5.5.$

Total H. P. $= 20 + 13.6 + 4 + 7.3 + 3 + 4 + 5.5 = 54.4.$

From the above one 70 H. P. engine should lift the span easily 100 ft. in 50 seconds with a wind-force being exerted of 30 lbs. per sq. ft.

Machinery.

The proportion of the gearing is such that the piston travel is to the motion of the bridge as 500 to 158. Under a boiler-pressure of 100 lbs. the force of one piston is sufficient to lift 20000 lbs. at the span.

Any part of the gearing can be readily removed and replaced in a short time. The friction-clutches used are of special design, and are operated by a wedge and toggle, which are in turn driven by a screw working in a stationary nut. The counterweights are in sixteen groups, each carried by a double cable. The water-counterbalance consists of tanks placed at convenient points and filled as required from a 2000-gallon tank located in one of the towers. Steam can be admitted in winter to prevent freezing.

A system of pawls arranged in a manner similar to a ship's

windlass provides means for slowly raising the span by hand-power. The signal device is also operated in much the same manner as on a ship.

A list of some of the advantages that the designer claims for the lift-bridge are given below.

Advantages of Lift-bridges.—The advantages of lift-bridges, in comparison with rotating drawbridges, are as follows:

1. A lift-bridge gives one wide channel for vessels instead of the two narrow ones afforded by a centre-pivoted swing-bridge.

2. There are no land damages in the case of a lift-bridge, as the whole structure is confined to the width of the street. These land damages in the case of some swing-bridges amount to a large percentage of the total cost of structure.

3. Vessels can lie at the docks close to a lift-bridge, which they cannot do in the case of a swing-bridge; consequently with the former the dock front can be made available for a much greater length between streets than it can with the latter.

4. The time of operation for a lift-bridge is about 30 per cent less than that for a corresponding swing-bridge.

The advantages of a lift-bridge in comparison with a bascule or a jack-knife draw, both of these being supposed to be without a centre pier, are as follows:

1. The lift-bridge can be made of any desired span, while in the case of the others the span is necessarily quite limited in length.

2. A lift-bridge can be paved, while the others cannot.

3. The lift-bridge is very much more rigid than any structure composed of two or more partially or wholly independent parts, a feature characteristic of the jack-knife bridge or the bascule without a centre pier.

4. In a lift-bridge the operating machinery is much more simple; and, in case that it should ever get out of order, the

span can be raised or lowered either by unbalancing or by simple hand mechanism, or by both combined.

The Van Buren Street Rolling Lift-bridge, Chicago (designed by William Scherzer). Cuts of this bridge are given on page 295.—This bridge has a clear opening of about 100 ft. There are two 21-ft. roadways and two 8-ft. walks. Each half of the bridge is operated independently, and the time required to open is about 35 seconds and to close 25 seconds. The wind-force was assumed at 18 lbs. per square foot on a surface of 3300 sq. ft., making a pull on the operating-strut of 165000 lbs. This is the stress for which the strut and the machinery are figured.

The adjustment in erection required considerable time and care to make the two halves of the bridge meet properly at the centre.

The bridge consists properly of three spans, the river span and the two anchor spans. The counterweights for balancing the cantilever arms are placed in the tail-girders, the amount required being about 129 tons, the exact amount required being determined after the bridge was in operation. The general arrangement of the floor-bracing, etc., is clearly shown on the cuts. The steel track-plate on the bottom of the segmental girder is 26 × 3 in., cut out to receive the teeth of the cast rack, which is secured to the masonry. The only connection between the two halves of the bridge at the centre is by means of a pin which is intended to take lateral stress only. The power is applied by means of a strut to which is attached a steel rack engaging with the rack-wheel. The operation of opening the bridge is as follows: The cam-crank at *A* (see Fig. 82) is revolved one quarter of a revolution and, acting on the cranks and levers shown, withdraws the pin-latch. A small roller at the rack-wheel strikes a cam which, acting on the levers, draws the tail-girder latch. The bridge is then free to move and the succeeding movement of the strut begins to lift the bridge.

Cast steel is used for the gears and all important castings.

There are two 50 H. P. Westinghouse motors on each half of the bridge.

The automatic brakes are 30 in. diameter and 6 in. face and are worked by compressed air. If at any time the current is cut off the brakes are automatically applied. The air for the use of the brakes and for operating the gates and signals is compressed by a pump driven by an eccentric on the end of the motor-shaft. An air-pressure of 35 lbs. is used.

Gates of the ordinary railroad type are used, and when once open the bridge itself acts as a gate.

The power used is a 500-volt current supplied by the power-station.

The average power required to operate the draw is about 60 H. P. for each side.

	Van Buren Street Bridge.	Wells Street Bridge.	Halsted Street Bridge.
Cost of substructure	$79600	$59000	$84700
" " superstructure	73100	86700	81400
" " machinery, engines, etc.	11150	4700	50000
	$163850	$150400	$216100

Tower Bridge, London.

Cost of substructure	$ 656000
" " steel	1685000
" " masonry	745000
" " machinery, engines, etc.	426000
" " miscellaneous	636000
	$4150000

The Tower Bridge has an opening of 200 against 100 for the Van Buren Street Bridge.

The weight of each leaf of the London bridge is 1000 gross tons, and the pressure used on the accumulators is 850 lbs. per square inch. Under a test load of 150 tons there was a deflection of $1\frac{5}{8}$ in.

When opening the Halsted Street Bridge in the quickest possible time 115 indicated H. P. was required; and at a

speed 20 per cent less the power developed was 96 H. P. The power furnished consists of two engines of 60 H. P. each using steam at 80 lbs. At the Van Buren Street Bridge the power required was 96 H. P. There are four motors of 50 H. P. each or 200 H. P. furnished. There seems to be doubt in the minds of many engineers as to the ability of the Van Buren Street Bridge being able to withstand the shock of a collision, which, in a crowded river like that at Chicago, is liable at any time. In this particular opinion favors the Halsted Street type of bridge. The use of motors gave a great economy in the amount of space required, and there has been no cause for dissatisfaction in the use of electricity as the motive power.

The Double Swing-bridge at Cleveland, Ohio.—There are two spans of 140 ft. each. The arms over the centre of the stream are the longer and are connected at their ends by a locking device. The assumed loads were 100 lbs. for the roadway and 80 lbs. for the sidewalks. The material used

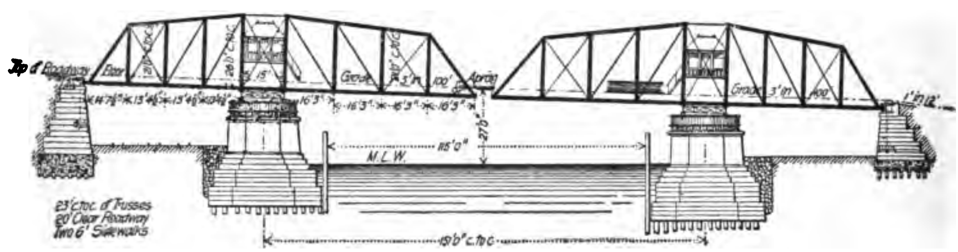


FIG. 44.
ELEVATION OF DOUBLE SWING-BRIDGE, CLEVELAND, OHIO.
Walter M. Rice, Engineer.

was soft and medium steel. Each span rests on a rim-bearing turntable with a drum 23 ft. in diameter and 3 ft. 3 in. deep. There are two distributing beams supported by spherical bearings on four cross-girders which in turn deliver the loads to the drum at eight points. The wheel-treads are made of rolled steel set in cement made of seven parts sulphur and one part coal-tar. Cast steel was used for wheels and pinions.

A 25 H. P. motor is used to turn each half of the bridge; an auxiliary 5 H. P. motor operates the air-compressors which drive the end-locking machinery. At the centre there is an apron 5 ft. 4 in. wide extending the whole width of the bridge. When the bridge is to be opened this apron is revolved by means of two steel plungers which pin the ends of the span together and which are in turn moved by the pistons of two $11\frac{1}{2} \times 19\frac{1}{2}$ in. air-cylinders. On counterweights attached to the aprons are racks which mesh with gears which are driven by the motion of the plungers. One movement of the piston in the air-cylinders draws the plungers and lifts the aprons.

The shore ends are locked by a toggle driven by air-pistons.

Hand-power is provided for use in emergencies.

The weight of the bridge is 428 tons.

The cost of the steel superstructure was $27000.

Electrical and compressed-air equipment, $5000.

Operating-houses, $592.

Wires, $500.

The substructure contained 2308 cu. yds. masonry, 447 cu. yds. concrete, 12223 lin. ft. piling, 4800 cu. yds. excavation, 2060 cu. yds. dredging, 127492 ft. B. M. (oak timber), 96452 ft. B. M. (pine timber), 46710 lbs. bolts and clamps; cost, $46000.

Designs for Drawbridge over Ship Canal at Duluth, Minn.—Fig. 45 shows the design made by I. A. McNicol. It

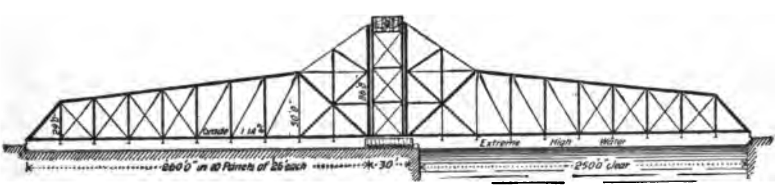


FIG. 45.

is a sliding draw of two equal arms which are entirely disconnected from each other and are supported by a central tower.

At the top of the centre tower there are tension-members which support the two arms, and which are provided with sliding nuts driven by machinery, by means of which the arms are lifted when the bridge is to be rolled back. The tower is proportioned to resist any unequal loading of the arms, wind-stresses, etc. All operating machinery is placed in a room in the tower.

The designer claims for this design the advantages of two independent spans when the bridge is closed. The strains are readily determined, and no platforms have to be raised and lowered to allow the arms to clear and to provide continuous floor. The time required to open the bridge is about five minutes, and the estimated cost of the bridge, exclusive of substructure, is $125000.

Fig. 46 shows the pontoon swing-span as submitted by Onward Bates and J. N. Warrington, as shown in the cuts below. One end is supported on an ordinary rim-bearing

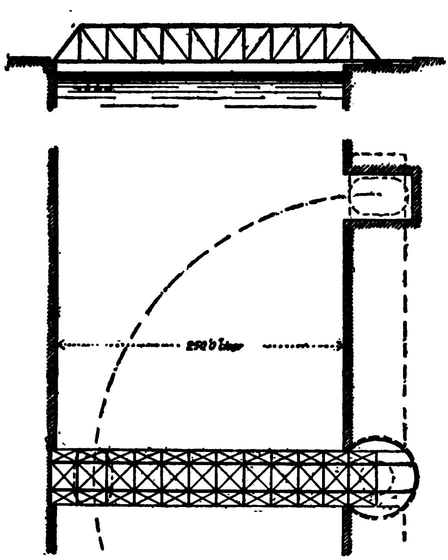


FIG. 46.
PONTOON SWING-BRIDGE.

centre or turntable and the other end upon a pontoon. When the bridge is to be opened the free end is lifted from its bearing on the masonry by pumping water out of the pontoon, and when clear the span is revolved about the pivot at fixed end by means of propeller screws working inside a hollow shaft in the pontoon. There is a hinge over the turntable centre which allows the truss to take a certain amount of vertical movement. A screw and engine of the capacity used on a large steam-tug are considered ample to operate the span. The time required to lift and turn is from one and one half to three minutes. The cost of this design is estimated to be $80000.

Figs. 47 and 48 are two designs submitted by J. A. L. Waddell. The first, Fig. 47, is a vertical-lift bridge of 260 ft. span. It can be raised to a height of 140 ft. above the surface of the water in the canal. There is a tower on each side of the stream about 170 ft. high. At the top of these towers are steel pulleys 15 ft. in diameter over which pass the

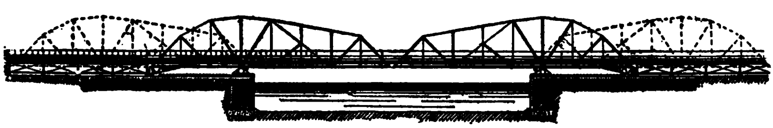


FIG. 47.

ropes or chains supporting the bridge proper. The cables are attached at one end to the trusses and at the other end to counterweights which exactly balance the weight of the span. The machinery has only to lift the weight due to snow, dirt, etc., provided the counterweights are properly adjusted. Two motors are used, and they are placed at the centre of the bridge. Steel shafts and gearing regulate the upward and downward motion. The total weight of the bridge and machinery is 1000000 lbs., and the time required to lift is from two to five minutes. Hand-operating machinery is pro-

vided in case accidents to the power devices render it necessary. The total cost of the structure is estimated at $125000.

The second design is a double sliding draw (see Fig. 48). To open the bridge each half is rolled back on steel rollers running on cast-iron tracks supported on the masonry. The trestle approaches are moved sideways far enough to clear the

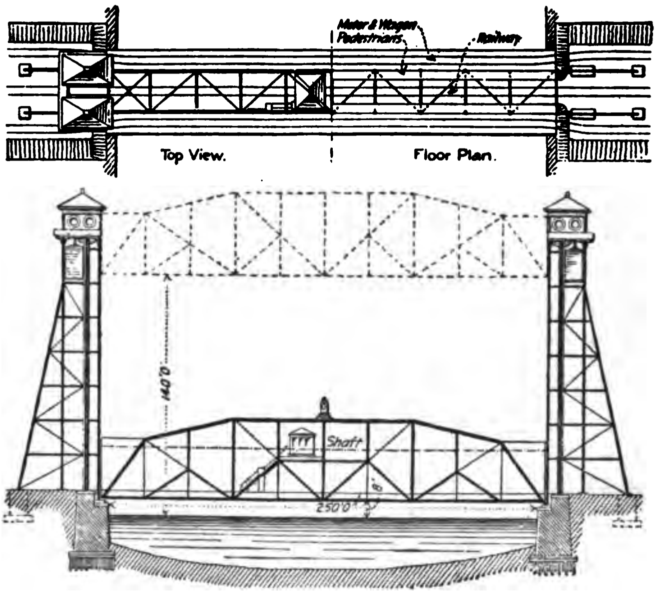


FIG. 48.

trusses of the draw-span as it rolls back. The shore arms are counterweighted to resist the weight due to unequal lengths. When closed the channel-arms are locked. Either steam or electric motors may operate the shafts, gears, and racks for performing the several movements. The time required to move the approaches and open the draw is about six minutes, and the estimated cost is $154000.

Fig. 49 shows the design presented by Messrs. Arenty & Sangdahl and which was awarded the prize. It is a single sliding draw, carrying street and railway traffic on two differ-

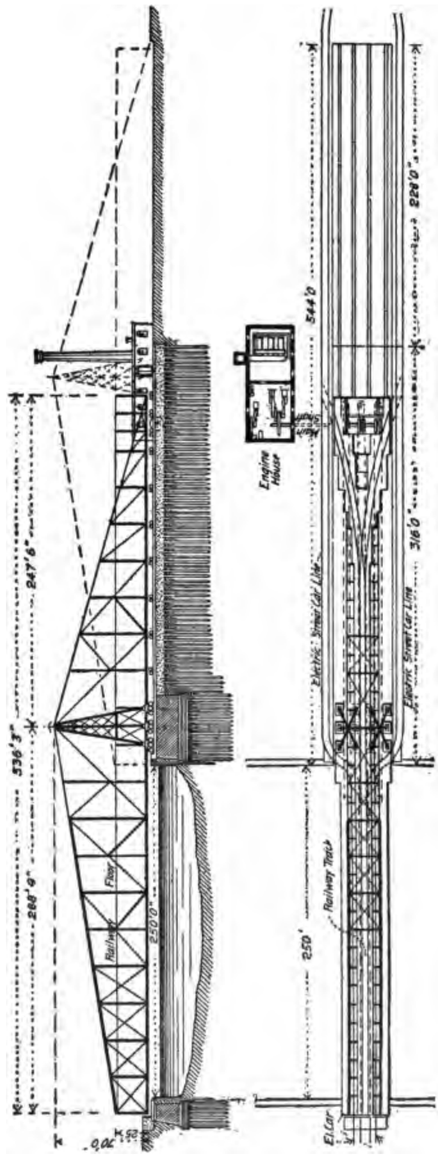


FIG. 49.
DULUTH ROLLING BRIDGE.

ent levels. The weight of the span is carried by the centre tower, which is thoroughly braced to give stability and which distributes the load over a large area. Under each of the twelve legs there is a truck supported on six double wheels which run on two 70-lb. rails. The rails rest on steel plates anchored to the masonry. The wheels are 3 ft. in diameter and have a 10-in. face. The shore arms of the trusses are supported at the panel-points by wheels 2 ft. diameter and 5 in. face which run on the same track as the wheels under the centre tower. By this means the trusses are made lighter and a considerable saving is made in the weights. A counterweight of 200 tons is carried on the three end-trucks. Toggle-joints placed in the tension-members next the centre tower raise and lower the channel-arms and release the tension when the bridge is closed. Six 1½-in. steel ropes arranged in pairs running over equalizing-sheaves 5 ft. in diameter and wound on drums 10 ft. in diameter and 30½ in. face are used to move the bridge. The engine is a 116 H. P. twin reversible engine, which is provided with powerful brakes always set unless released by the engineer. A separate regulating- and reversing-engine is used to give an accelerated motion at the beginning and a retarded motion at the end of the run. The main engines have 28-in. cylinders and 48-in. stroke. There is also in the engine-room a dynamo operated by an engine with 10-in. cylinders and 24-in. stroke. The weight of the bridge is 2400000 lbs., time of opening or closing one minute, and estimated cost $236000.

Sooysmith & Co. submitted a double sliding draw (Fig. 50). Large pins made of phosphor-bronze couple the two arms together at the centre when the bridge is closed. Engines located under the centre of each half of the bridge actuate screw-gearing which drives the pins into tight-fitting holes in heavy clamps attached to the chords. To open the bridge the pins are first drawn; each half of the structure is then revolved in a vertical plane about the large pins placed

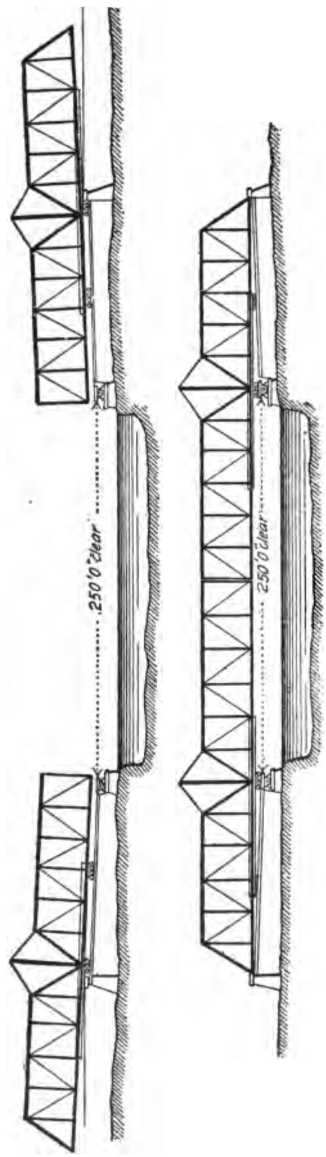


FIG. 50.
DULUTH DRAWBRIDGE.

below the nests of rollers at the bulkheads until the shore ends are raised high enough to clear the permanent roadway. Winding-engines and cables passing over sheaves fixed beneath each end of the bridge then pull the two arms back until the channel is clear. The rollers on which the bridge rests are shown in the diagrams.

The track on which the rollers run rises about two feet between the bulkhead and the end of the structure. The shore arms of the bridge have a rising grade of one per cent; between the bulkheads the grade is level. By this means the moving parts clear the tracks and the roadway at the point where the fixed and moving parts join. Heavy guides prevent the rollers from getting out of line and also keep the moving arms in their proper alignment during the movement. When closed the ends are locked to the fixed structure. Each half of the structure is counterweighted in such a manner that the arms are always balanced and the machinery has only to overcome the friction. The estimated cost of the structure was about $180000.

One of the advantages claimed was the small amount of space required and the fact that no land is required outside the right-of-way lines.

The design of Mr. J. N. Ostrom (Fig. 51) is a double lift, each of the two arms being 127 ft. long and the fixed

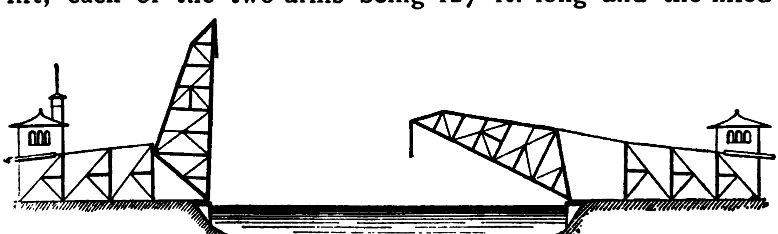


FIG. 51.

support at each end 95 ft. long. Provision is made for a single-track railroad, two electric-railroad tracks, and a roadway 20 ft. wide, with a 5-ft. walk at each side. The channel-

arms are hinged at the abutments and at the centre are supported by swinging bents or piers which take a bearing on pedestals built in the bed of the stream. The span is opened by revolving the arms upward on the abutment-hinges, hydraulic machinery for this purpose being placed in the shore arms or supports. The arms descend by gravity, the movement being checked and controlled by the machinery. There are two sets of hydraulic cylinders, three cylinders to a set, and two sets of air-accumulators. One set of boilers, pumps, etc., operates both sets of cylinders and accumulators. The pistons of the cylinders are attached to moving girders which in turn connect to the walking-beams which are coupled to the channel-arms of the bridge. Two 50 H. P. boilers supply power to two compound duplex pumps, one of these pumps operating each arm. If necessary they may be combined and the whole power applied to one arm. The arms can thus be operated singly or together.

The time required to open or close the bridge is from 1½ to 3 minutes. Two men operate the entire machinery. The estimated cost is $125000.

The Lift Bridge over the Sixteenth Street Viaduct, Milwaukee, Wis.—The bridge (Fig. 52) has a 40-ft. roadway, two 6½-ft. walks, and a total width of 53 ft. inside hand-

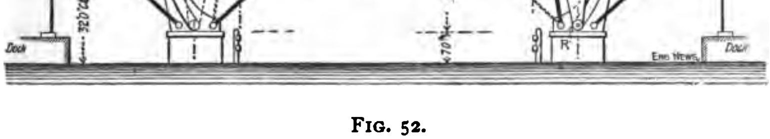


FIG. 52.

rails. The roadway has a crown of 2 in. The floor is made of one layer of 3-in. and one of 2-in. plank. A live load of

100 lbs. on the roadway and 80 lbs. on the walks was assumed in designing the girders, and the beams and stringers were proportioned for a 16-ton electric car on each track. The dead weight of each moving half is 130000 lbs. The main features of the design are shown in Fig. 39. The curve of the track is such that as the bridge swings up the resultant of the weight is always perpendicular to the tangent of the curve, and hence no part of the weight has to be lifted and the frictional resistances alone have to be overcome in operating the bridge. The arm is not balanced except under uniform load, and for a concentrated load the roller end of the girder must be latched to prevent motion either up or down. The friction of the shaft is reduced by a set of steel spring-rollers arranged around it at the bearings. This arrangement also tends to relieve any slight irregularity in the track. The operating machinery consists of two struts attached to the main girders at the centre of gravity of the swinging portion; steel racks are fastened to these struts and are driven by pinions which are keyed to the main shaft. The shaft is turned by a worm and worm-wheel which are in turn driven by the motor. A hand-turning gear is also provided. To lessen shocks in closing the bridge a strong spiral spring and a lever are attached to the operating-strut and as the bridge closes this spring is compressed and absorbs the surplus power. The bridge has been opened in 15 seconds with 11 H. P. Ordinarily it is opened in 30 seconds. The cost of the bridge complete was about $42000. The advantages claimed for it are:

1. It permits a floor of the heaviest kind, with paving, or of asphalt and concrete on buckle-plates, without making the bridge too dangerous to operate, since there is perfect balance of moving portion at all times and seasons.

2. On account of the peculiar motion of the swinging portion the effects of wind-forces are much reduced and can easily be controlled.

3. The central deflection is small, due to the supporting swinging struts.

4. Both ends of streets or approaches are closed by the bridge floor when the bridge is opened.

5. It requires no centre pier and permits a single wide channel for vessels.

6. This construction requires no towers projecting above the roadway and obstructing the view of the river.

Cuts of this bridge are shown in Figs. 84 to 84*c*.

Newtown Creek Draw.—Fig. 53 shows the design submitted by Mr. Brown. It is a hinged-lift structure in two halves, supported by towers on each shore and operated by hydraulic power. More in detail the design is described by Mr. Brown, as follows:

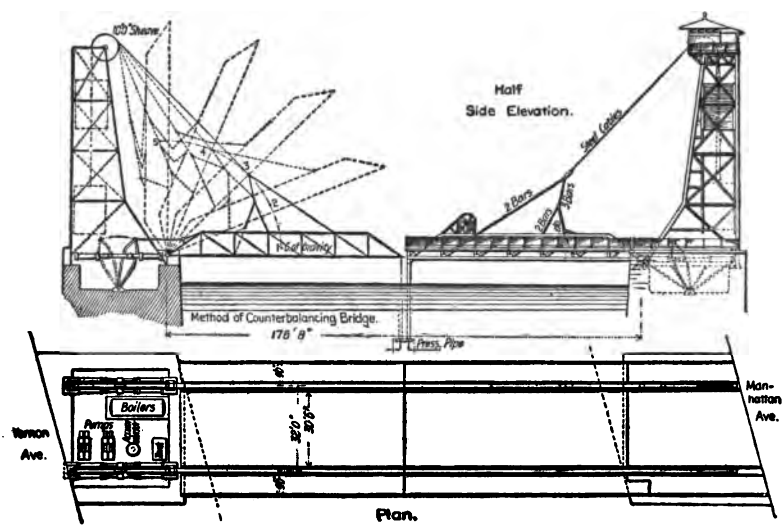


FIG. 53.

NEWTOWN CREEK BRIDGE.

Design of Thomas E. Brown, C.E.

"The bridge consists of two hinged spans 80 ft. 4 in. long from centre of hinge to centre of river, or a total of 176 ft.

8 in. between hinges, and two towers about 74 ft. high, surmounted by houses for the operators.

"The outer ends of each span will rest on steel trestle-bents hinged to the lower chords of the trusses. These will rest on castings in a sub-pier at bottom of river. The trestle-bents will rise with the bridge and fold parallel to the floor of the spans, leaving the full depth and 150 ft. width of stream clear of obstruction. This motion will be controlled by a positive automatic connection, independently of the weight of the bents.

"Small hydraulic pressure-pipes will be laid, as indicated, to the central sub-pier. Any silt drifting under the feet may be washed away by turning pressure on these pipes. The bents may be dispensed with if preferred, and the span supported by suspenders from the towers, as indicated on the plans.* This will increase the cost of the bridge.

"Each span will be balanced by two cast-iron weights in steel frames running in guides in the framework of the towers. These will be connected with the span by twelve 1½-in. steel cables (six on each side) passing over two 10-ft. diameter steel-riveted wheels placed on top of the tower. The weights will be subdivided and the cables attached to them by equalizers with adjustable connections.

"The cables will be connected to the spans by a system of links, so arranged that the counterweight, though constant, will balance the span in all positions, except near each end of the travel, where the position of the links is such as to cause the weight to act against the moving span and assist in bringing it to rest. This arrangement is shown by an explanatory diagram on the drawings.

"On the ends of the spans are placed curved lattice buffers which will engage the cables should the spans pass the perpendicular line. These are so arranged that a wind-

* *Engineering Record.*

pressure of over 30 lbs. per sq. ft. will not be sufficient to force spans against towers.

"Each span will be operated by eight hydraulic rams, 12 in. in diameter and about 8 ft. stroke, four for lifting and four for lowering. These will be supplied with pressure from a compound duplex pump. An additional duplex pump is provided to act as a relief for the compound pump and to increase the working pressure during high longitudinal winds. These will be automatically controlled by variable-power hydraulic valves.

"All hydraulic pipes, fittings, and cylinders will be tested to 2500 lbs. per square inch, and will be so duplicated that the bridge will be capable of operation in the event of disabling of any two of either set of four rams or their connections. The rams are of sufficient size and strength to resist a longitudinal wind-pressure of 30 lbs. per square foot on the entire floor of the bridge. The arrangement of pumps is such that a range of hydraulic pressures may be obtained up to 1200 lbs. per square inch. Under average conditions the working pressure will be about 250 lbs. per square inch. Mechanical automatic stops will be provided in connection with the operating-valves which will bring the bridge to rest gradually at the end of its travel. Hydraulic buffers will also be provided on the ends of the rams, for the same purpose, as an additional safeguard.

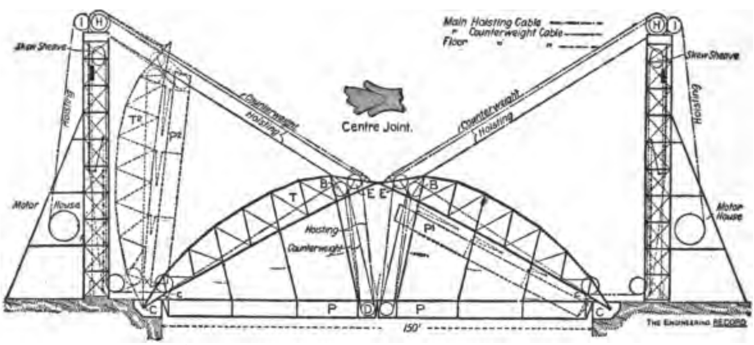
"Hydraulic speed-governors will be provided, by means of which the speed of the bridge will be controlled or limited independently of the operator. The operator may move the bridge as slowly as he may desire or stop it entirely at any point of its travel, but he cannot, nor can a puff of wind, increase the speed beyond the limit at which the governors are set.

"The bridge may be operated in calm weather in 30 seconds and closed in the same length of time. Under average conditions the time would be about 45 seconds open-

ing and 45 seconds in closing. The bridge may be operated directly by a man in the tower, or by signal-bell to a man in the engine-room, as may be preferred.

"The plan includes suitable boiler and appurtenances, and the entire power apparatus is located in the foundations of the tower. If a supply of electric current is available electric motors may be substituted for hydraulic rams and the boiler be dispensed with. We prefer a hydraulic plant for this class of work, considering it more reliable and certain in action, and less liable to derangement."

Fig. 54 shows a half-span of a structure designed by Mr. W. H. Breithaupt. It is a three-hinged arch, made in halves,



DESIGN FOR A BRIDGE OVER NEWTOWN CREEK, BROOKLYN, N. Y.
MR. W. H. BREITHAUPT, M. AM. SOC. C. E., ENGINEER.

FIG. 54.

with the bridge floor suspended from it and hinged to it. The following is from the description submitted by Mr. Breithaupt with his design:

"The floor is in two parts and is hinged to the arch near its ends. The arch is made to part at the centre point, and each half is arranged to revolve around its end-pins. There are two towers, one at each end of the arch. Over the tops of the towers, and attached to the two halves of the arch at the centre joint, are cables, by means of which the mid-stream ends of the floor and arch halves respectively are raised. The floor-suspenders are stiffened pairs of eyebars, jointed at the

centre of their length so as to fold between the floor and the arch as the floor is raised. In opening the bridge the floor halves are first pulled up against the arch; then the arch parts, with the floor parts close against them, are raised to approach a vertical position against the towers. In many cases, such as to allow passage of single tugs or barges having no high projections, it will be necessary to raise the floor parts only, without disturbing the arch. When the bridge is open the roadway from either side is shut off by the bridge floor standing on end across it and forming an effective barrier.

"There are two sets of counterweights, one for the floor and a main set for the entire bridge. The floor counterweights are of constant counterbalance, consist of a single weight each, and operate, with the exception of a short-leverage pull afterwards, only in raising the floor against the arch. The main counterweights consist each of a chain of weights suspended from the respective cables and having its lower end attached to a fixed point. As the counterweight descends it gradually reverses and its weight is gradually and evenly, without shock of any kind, transferred to hang from the fixed point spoken of. This arrangement provides for uniform counterbalance as the arch parts rise.

"The bridge is operated by means of electric motors, one in each tower, connected by a cable under water and controlled from one switch by a single operator. The mechanism is so arranged that the two arms of the bridge move together and that one cannot get ahead of the other. The motors connect to the drum-shafts by means of worm-gearing, as an additional safeguard in regulating the movement of the bridge-arms. The motive power may also be a steam-engine in each tower, electrically controlled to operate synchronously. A hand-winding arrangement is provided at each worm-shaft, so that the bridge can be operated by hand-power in the very improbable case of this being necessary.

"The floor can be raised to the arch in less than 20 seconds and the full raising of the bridge can be done in 40 seconds. No locking of any kind is required on closing, and therefore no time is taken up in this way.

"Forming the centre joint of the arch, one arm has a convex end and the other a corresponding cavity with guide projections, which serve to bring the two arms to their proper bearings against each other. The surface of contact between the arms is, in vertical section, a little less than a vertical semicircle, a slight revolution of one end about the other being provided for. This allows for expansion and contraction of the arch-arms, without disarrangement of any part. To avoid shock on the coming together of the two arms hydraulic buffers are interposed at the tops of the towers to check the main counterweights as they come to their upper limit of travel. Buffers are also placed at the points where the floor impinges against the arch during the raising of the bridge."

A 50 H. P. motor in each tower will open the bridge in 40 seconds.

The New Rock Island Bridge at Davenport.—The bridge was designed by Mr. Ralph Modjeski, who was also the chief engineer in charge.

The bridge is a double-track railroad, and highway, with a total length of 1850 ft., including a draw-span of 366 ft. operated by electric power. The roadway floor is 26 ft. wide, with a head-room of 12 ft. 6 in., and the railroad floor 26 ft. wide, with a head-room of 21 ft., or about 5 ft. more than in the old bridge. This floor is made up of four lines of stringers 6 ft. 3 in. centre to centre, and is covered with a corrugated-steel floor 26 ft. wide, which, being water-tight, acts as a roof for the protection of the roadway beneath.

The operating machinery consists of five different devices, viz., the swinging machinery, the rail-locks, the end-jacks, the interlocking and controlling system, and the emergency hand devices. The swinging is done by means of a cast-steel

rack attached to the tread on the centre pier and having sprocket-teeth. Two vertical shafts, one on either side of the bridge, supported by brackets and fitted with cast-steel sprocket-wheels at the lower end, transmit the power to the steel rack by means of an endless chain carried by the sprocket-wheels and engaging in the teeth of the rack. On the upper end of these vertical shafts are sprocket-wheels, which are connected by chain to a vertical driving-shaft midway between them, which rises to the floor of the machinery-room. An interesting feature of this part of the mechanism consists in the fact that all of the vertical shafts are run on ball step-bearings. By means of a train of gears a 50 H. P. electric motor is connected with the shafts, thereby transmitting the power to the rack below.

The rail-locks consist of heavy steel slides fitted to the outside of the rails on the bridge with those on the fixed span, thus giving a continuous line of rail for the passage of the wheels. The slides are operated by means of a pneumatic cylinder located in the centre of the bridge, all four slides at each end of the bridge, and both ends, moving simultaneously.

The end-jacks are of the semi-toggle type, consisting of two parallel pairs of bars attached to the end-beams, directly under the chords, by pins, so as to turn freely, while on the lower end are rollers which rest on bearing-plates on the shore abutment. By means of a pneumatic cylinder, a centre crank, and struts connected to the roller-pins the jacks are forced to a vertical position when the bridge is closed, and are drawn to release the bridge when it is to be opened. For the interlocking a Hall signal is placed on each of the fixed spans, within a few feet of the ends of the draw, standing normally at danger. Connected to each of the jacks and rail-locks are electric switch-boxes, which are also connected by wires to an indicator in the machinery-room. When any one of the jacks or rail-locks is in a closed position (or when all are) a red lamp is lighted in the indicator (one lamp for each jack

or lock), and when the bridge is released to swing a white light is shown, replacing the red. The man in charge can set the signal to safety only when the end-jacks and locks are set. Should, for any reason, the bridge not be properly locked the engineer cannot receive his signal to enter upon the draw.

The controlling device makes it impossible for the operator to swing the bridge until first the rail-locks and then the end-jacks have been released. Emergency hand devices are provided, by which the bridge can be swung in case, for any reason, the mechanism becomes inoperative. An air-compressor, driven by an electric motor, is located in the machinery-room, supplying power for the various cylinders.

In calm weather 27 H. P. will turn the draw. The bridge works very satisfactorily. As in all chain-gearing, there is a small amount of lost motion which interferes to some extent with the smooth operation of the machinery. The friction is considerably less than with the ordinary rack and pinion. Views of this bridge are given in photographs on pages 304 to 306.

The 520-ft. Swing-span Interstate Bridge, Omaha, Neb.—There are two trusses 520 ft. long centre to centre of end-pins, 95 ft. high at the centre, 50 ft. high at the first hips, 25 ft. high at the end hips, and 30 ft. apart centre to centre. Between the trusses are carried the two railway-tracks, and on the outside cantilever-brackets are the two motor-tracks, roadways and the sidewalks, all being at the same level. Each truss consists of two parts linked to the centre tower resting on the turntable. The loads for which the bridge is calculated are as follows: A live load consisting of the train load (which is Waddell's standard engine-loading Class X), and 100 lbs. per square foot of roadways and sidewalks. This load is used in computing the floor-systems and primary truss-members. The webs and chords are computed for a live load assumed at 9600 per linear foot, one arm loaded, or 8000 lbs. per linear foot, both arms loaded. The

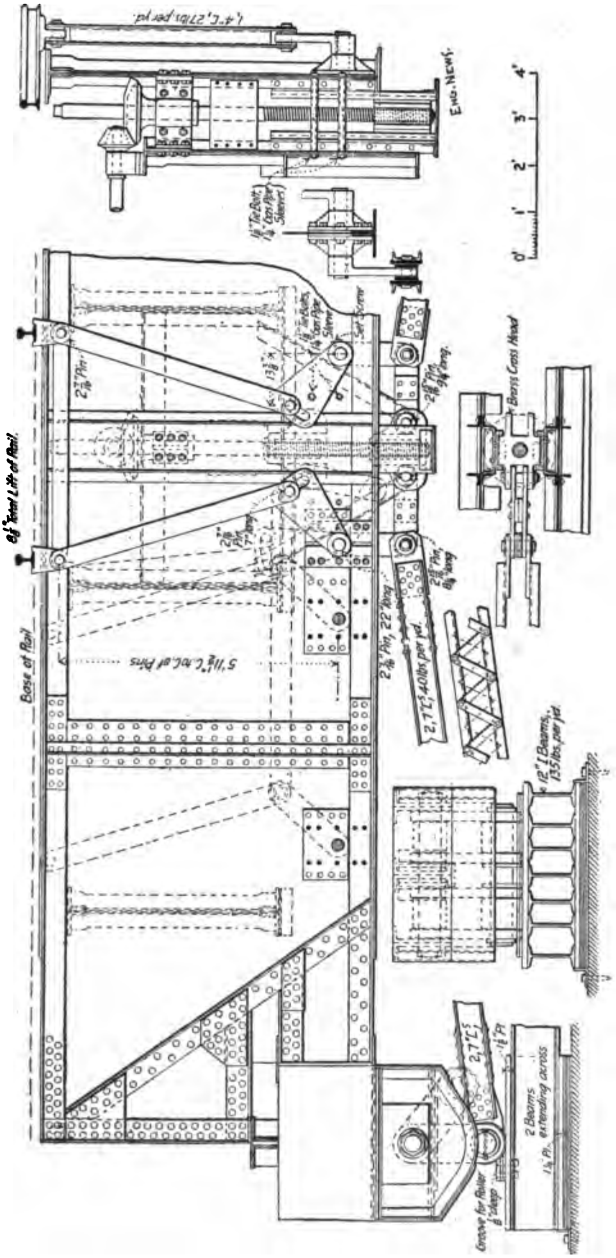


FIG. 55.
SWING-SPAN INTERSTATE BRIDGE, OMAHA, NEB.
J. A. L. Waddell, Chief Engineer.

wind-pressure is taken at 600 lbs. per linear foot for the lower lateral system, and 280 lbs. per linear foot for the upper lateral system. The dead load is assumed at 6100 lbs. per linear foot.

The bridge is operated by two 40 H. P. Waddell-Entz motors, capable of running at 270 revolutions per minute, placed on the floor of the engine-room on the first horizontal tower-girders. For turning the bridge the motors gear with a horizontal shaft which in turn gears with an equalizer on a second horizontal shaft which gears at its ends with the vertical shafts to the drum. For lifting the ends the motors gear on the opposite side with a horizontal shaft which in turn gears with the vertical shaft to the drum. It will be seen that the end-lifts are operated by a horizontal shaft running lengthwise of the span and operating a screw.

The Michigan Avenue Bascule Bridge, Buffalo.—The erection of the new bridge was accomplished entirely without false work and without interfering with navigation.

The trusses of the swing-span, 77 ft. in length and 9 ft. high, were shipped in one piece, and the big tower-legs, 75 ft. high and weighing 17 tons each, in the same manner.

A large frame bent 56 ft. in height, running on the sidewalks of the tower span, was employed. In the erection of the tower spans, counterweight-tracks, etc., the ironwork of the swinging-span, which hinges at the foot of the tower by means of 8-in. pins in bronze bearings, was put together and the oak floor laid while the trusses were in a vertical position.

The photograph on page 302 shows the general appearance of the bridge in open position.

The live-load ties consist of two $6 \times 1\frac{1}{8}$ in. eyebars laced and hinged at ends and centre.

In order to accommodate the length of bars when the swinging-span is in an upright position the pin P at upper end of live-load ties is allowed a vertical motion of about 2 ft.,

and is arranged so that it is locked automatically when the bridge is in a closed position.

This arrangement, in connection with the back-strut *B*, holds the live-load ties in position and effectually prevents excessive vibrations, which were feared on account of the long overhanging arms. The back-strut is provided with a locking-wedge *W* at its rear end which, when bridge is down, is thrown into position by a single hand-lever in the engine-house.

The dead weight of each moving span is 60 tons, which is carried by the dead-load ties, consisting on each side of two eyebars about 30 ft. in length, to which the four counter-weight-cables are connected by an equalizing device. The $1\frac{1}{2}$-in. crucible-steel cables are 62 ft. long, have hemp centres, and are provided at each end with ordinary babbitted socket-clevises. They pass over two double-groove sheaves, keyed to a 6-in. shaft at top of tower, and are fastened to the shaft of the counterweights by means of a connecting hanger.

The total counterweight of one side is 70 tons; each one-half, or 35 tons, is made up of ten large castings and of six smaller ones assembled upon a 6-in. shaft 8 ft. long. The ten main castings are 6 ft. 6 in. in diameter and 5 to 7 in. thick. Two of them are grooved to run upon the curved counterweight-track. This track consists of two 60-lb. T rails spaced 3 ft. centre to centre and supported by girders. A small adjustment is provided for by means of shimming-plates under the curved rails. Calculations to determine the correct shape of the curved track were quite tedious, owing to the fact that the back-strut, live-load ties, varying length and direction of the cables, etc., had to be taken into consideration. The fixed spans at shore ends are anchored against uplift by means of long anchor-rods built into the masonry. The power for operating the bridge is applied to the moving spans at the hip through the operating-strut by means of a triple

screw about 18 ft. long and 6 in. in diameter; it has $\frac{3}{4} \times \frac{3}{4}$ in. thread, $4\frac{1}{2}$-in. pitch, and requires 46 revolutions, or a stroke of 15 ft. $3\frac{1}{2}$ in., to open the bridge. The screw itself is held in position by means of a thrust-box. This is a bronze casting with grooves to receive the collars turned on the screw-shaft. The bronze nut through which motion is conveyed to the operating-strut is mounted on a travelling carriage, with top and bottom tracks, to take up vertical reaction when the operating-strut is not working in a horizontal direction.

Near the thrust-box a bevel-gear is arranged to slide upon a feather in order to connect and disconnect the hand-gear, the vertical shafts of which are carried down to a cross-shaft underneath the roadway floor. A $4\frac{1}{2}$-in. shaft extends from the coupling to the sprocket. This sprocket has 15 teeth.

The sprocket-chain is a special chain. It is intended to transmit a strain of 7000 lbs. at a speed of 250 ft. per minute. The upper sprocket-wheel of 8 teeth is mounted on a jack-shaft, to which power from the engine is communicated by means of a lighter chain, capable of transmitting 1600 lbs. at a speed of about 900 ft. per minute.

Both link-chains run on a double sprocket-wheel upon the engine-shaft. A jaw-clutch working on a feather upon the shaft throws engine in and out of gear. The engine itself is a double-oscillating reversible Kriebel engine, with $7\frac{1}{2} \times 9$ in. cylinders and speed of 300 to 350 revolutions per minute.

With a boiler-pressure of 50 lbs. it is capable of opening or closing the bridge in 45 seconds or less. The bridge, so far, has not been severely tested by wind-pressure; but, since the boilers are built to carry 150 lbs. steam-pressure, while only 50 lbs. is required to operate the bridge under ordinary conditions, it is believed that ample power is in reserve to operate the bridge during a moderately high wind.

On account of requiring less attention hard coal is burned under the boilers, which are of the submerged-tube type 4 ft. in diameter, 10 ft. high. The engine-houses are conveniently

arranged with coal-bin, coal-hoist, water-supply, electric lights, etc.

Both halves of the bridge are practically of identical construction. It must be opened 40 to 50 times a day for the passage of vessels.

Team and foot travel across the bridge is very heavy. A considerable shear and vibration was expected where the two halves meet at mid-stream on account of the very long swinging-trusses; however, under very heavy moving loads it only reached a maximum vertical or horizontal motion of $\frac{1}{2}$ in. A centre lock was hardly considered necessary, but finally a small contrivance consisting of a 3-in. square pin at each truss which could be thrown in and out by a hand-lever was provided. This device effectually prevents shear and lateral motion and is worked by the bridge-tender, who is in control of the structure and who signals the operators in engine-houses at each side when the bridge is to be opened or closed.

So far the bridge has worked in a very satisfactory manner, and it is believed that with ordinary good care it will prove to be a reliable structure.

A Recent Example of Automatic Interlocking Machinery for Operating a Swing-bridge.—The interlocking and automatic devices used on a long span recently constructed by the Edge Moor Bridge Works are shown in Figs. 56 and 57. There are three sets of friction-clutches operated by the levers at *A*, *B*, and *C*. At *Y* there is a set of equalizing-gears for distributing the load equally to the two pinions. Fig. 56 shows the bridge closed, with all the levers locked, with the exception of lever *A*, which can be thrown to the right. The action briefly described is as follows:

Assume the draw closed, the ends lifted, and the signals showing a clear track. If the bridge is now to be opened the first operation will be to throw the signals and draw the rail-clamps. By throwing lever *A*, which controls the clutches

S and *T* to the right, clutch *T* is engaged and drives the line of machinery, which releases the danger-signals at the end of the span and at points 400 ft. distant. These signals are

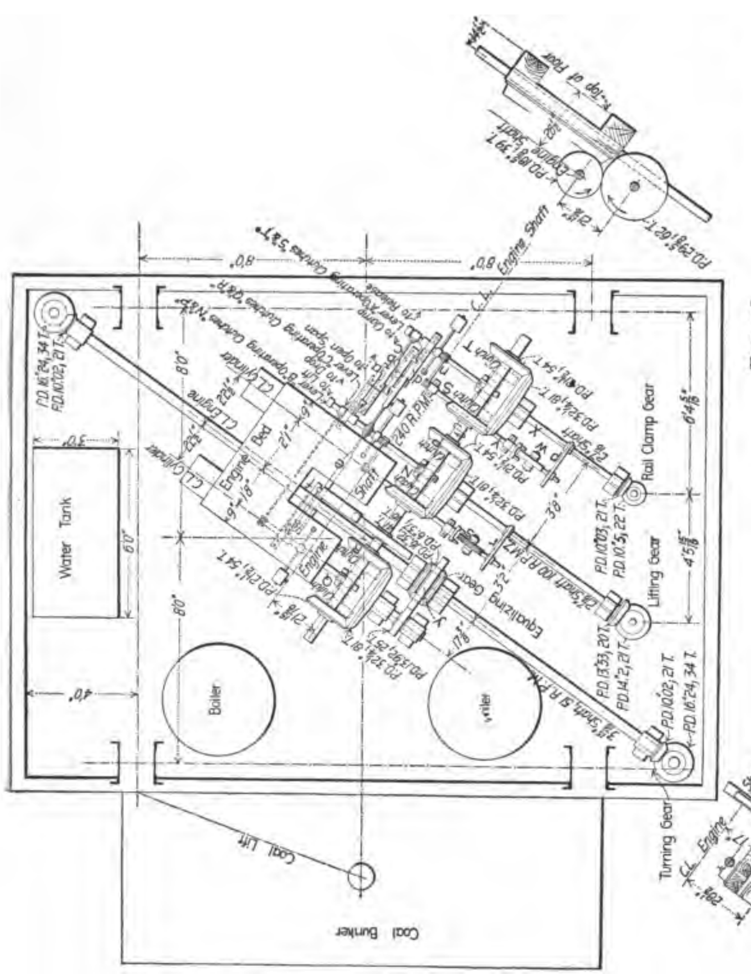


FIG. 56.
ARRANGEMENT OF AUTOMATIC INTERLOCKING OPERATING MACHINERY FOR A SWING-BRIDGE FOR THE WHEELING & LAKE ERIE RAILROAD.
John N. Ostrom, Consulting Engineer.
Edge Moor Bridge Works, Wilmington, Del., Builders.

thrown by springs or counterweights, which are released by the withdrawing of an arm which upon the return movement revolves the vertical shaft to which the signals are attached

and compresses the springs. At the same time that the signals are thrown to danger the rail-clamps are drawn back

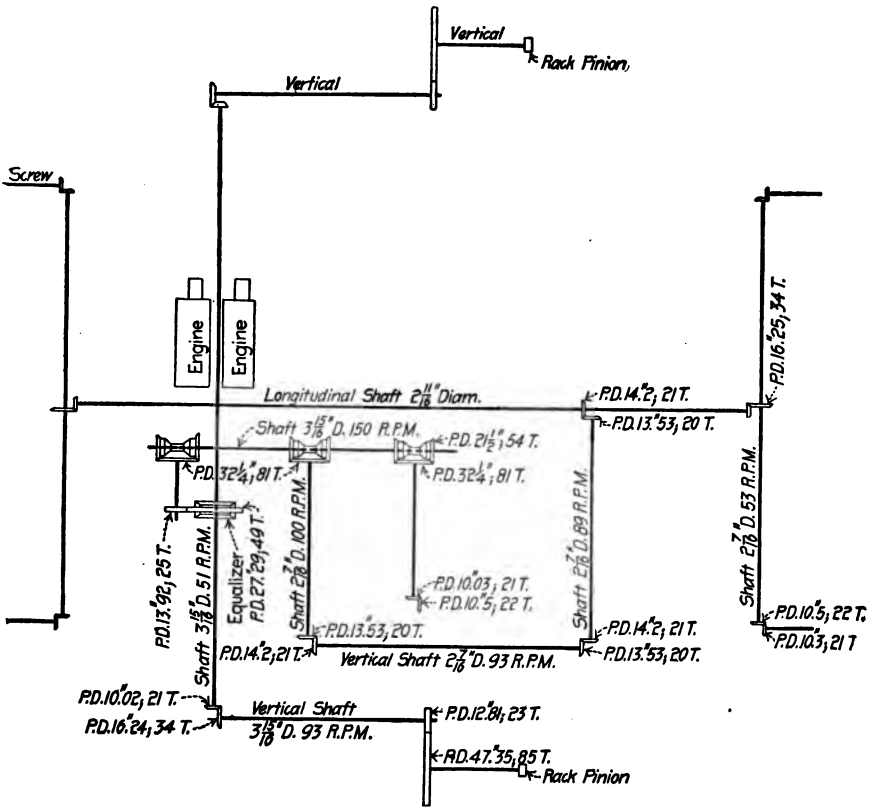


FIG. 57.

DIAGRAM SHOWING SIZES OF SHAFTING AND GEARING LEADING FROM ENGINE TO POINTS WHERE POWER IS USED.

by a system of gears and levers, controlled by the same shaft which operates the signals.

When the lever *A* was thrown to the right it pulled the lock-bar *a* to a position where it unlocked the lever *B*, which operates the end-lifts. The bar *b*, however, still locks this

lever. The lever *C* is also locked, so that the turning machinery cannot be moved in either direction. After shaft *X* has revolved a certain time, sufficient to draw the rail-clamps, the chain-gear *p* has turned the worm-wheel *w* until the projection shown on the right-hand side of the wheel strikes the arm on the fork, which is attached to shaft *Y*, and by revolving this shaft throws the rocker *n* and brings the lever *A* to a vertical position again, which frees clutch *T* and stops this portion of the machinery. As the lever *A* moves back to the vertical position it brings the notch in lock-bar *b* opposite the lug on lever *B*, and thus releases this lever. Lever *B* can now be thrown to the right, but is still locked to the left. If *B* is now thrown to the right clutch *P* is engaged and the end-lifts are drawn. This motion of *B* throws the lock-bars *c* and *d*, which lock *A*, and also throws *e* and *f*, *f* being thrown so as to unlock lever *c*, while the bar *e* still locks it. Lever *A* is now locked in one direction by the bar *c*, and in the other by the worm-wheel and rocker *n*. Lever *C* is also still locked. When the shaft *Z* has revolved a sufficient time to draw the end-lifts the chain-gearing and the worm-wheels (similar to the arrangement on shaft *X*) throw the clutch *P* out of gear and bring the lever *B* to a vertical position again. As *B* moves to the vertical position it throws the bar *e*, which unlocks lever *C*. This lever can now be thrown in either direction for turning the draw.

Lever *A* is meanwhile locked in both directions and *B* in one. In closing the draw the operation just described is reversed. There are indicators which show the progress of the different operations.

Impact-wheels.—Where the requisite pressure can be secured the impact water-wheel would seem to furnish a cheap and satisfactory power for drawbridge use.

A 250-ft. draw on the Southern Pacific Railroad is operated by a Pelton wheel. A 24-in. wheel is used and the available head of water is 80 ft.

The speed of the motor is 342 revolutions per minute and of the turning-shaft 4.1 revolutions per minute. When at full

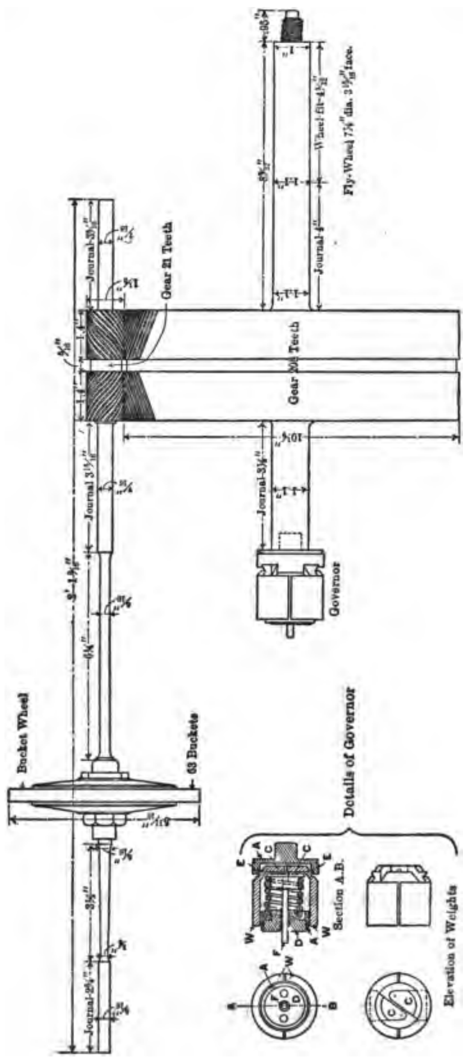


FIG. 58.
IMPACT-WHEEL.

speed the motor develops 8 H. P. With a half head of water and the motor making 75 revolutions per minute 2 minutes

and 12 seconds are required to unlock the ends and turn the draw.

Fig. 58 shows the general arrangement of an impact-wheel.

Eight-track Railroad Drawbridge over the Chicago Canal.—This bridge which is soon to be built over the canal in Chicago is in some respects a remarkable structure. The clear span required in the instructions to bidders is 120 ft., if a single span be used. If two spans are employed, each must have a clear opening of 80 ft. Among the designs submitted are the horizontal rolling and several styles of rolling lift or bascule. There are four sections to the bridge, each section carrying two tracks. This is rendered necessary by the sharp angle which the centre line of the bridge makes with the abutment lines (the skew). The total width of the bridge is about 115 ft. The Schenke type, similar to the Sixteenth St. bridge in Milwaukee and which is illustrated on page 299, was finally decided upon by the committee. Mr. C. L. Strobel submitted a design which was favorably considered, but as it required a centre pier it was not adopted, as the committee considered the use of the centre pier unnecessary, and a decidedly objectionable feature in this narrow stream. There can be no doubt, however, that the action of the bridge under railway traffic would be more satisfactory if a design using the centre pier had been adopted.

The cost of some of the designs submitted are give below:

	Without machinery.	Including machinery.
Design of C. L. Strobel........	$235424	$337424
" M. Schenke.........	366759	416750
" M. Scherzer.........	369140	510880

Description of Plates.—Plates A and B illustrate a single-track railroad swing-draw, provided with hand-turning machinery, and end wedges driven by a worm-screw and levers which are connected by rods running along the side of

the bridge. For moderate-length spans this method is believed to be preferable to shafting and gears.

All joints are riveted and there are no adjustable members in the truss. Stiff bracing latticed is used in the plane of the top chords except at centre, and the bottom laterals attach to the stringers at intersections. Bracing is also used between the stringers. The dead load is entirely carried by the centre pivot, and the live load by wedges below the trusses at the centre and at the ends. The end wedges lift the ends sufficiently to prevent hammer under partial loads and the centre wedges are just driven to a bearing. The centre pivot has disks made of hardened steel and phosphor-bronze. Such a bearing will safely carry a load of 5000 lbs. per square inch for dead loads and 2500 lbs. per square inch for live loads.

The bridge has given excellent service and is believed to represent the best modern practice.

Plates C, D, E, and F illustrate a 290-ft. railroad draw, with pin-connections and no adjustable members in the truss. The machinery of this span has already been discussed. Both hand and engine operating machinery are used. In the engine-house there is an indicator which shows the operator the position of the end wedges. If the engine is not stopped or the machinery thrown out of gear when the wedges are driven the eccentric merely continues to throw them in and out without injury to any part of the machinery. As a precaution there is a foot-brake attached to the main shaft in the engine-house. It has been found, however, that the friction-clutches which control this shaft are an efficient brake in themselves. The engine is a 12 H. P. gas-engine running at about 325 revolutions per minute. The end-latch is worked independently by a lever placed near the engine. The rack, pinion, and most important gears are of cast steel. The wheels and less important gears are cast iron. The load on the wheels is say 416000 lbs., and there are 38 wheels 12 in. diameter and 6 in. face. The load per lineal inch is 1820 lbs.

(see page 61 of "Plate-girder Draw-spans"). The bearings for pinion-shafts are well supported, and there is little spring or jar as the teeth take their bearing. There are eight points at which the drum receives its load.

Plates G, H, I, and J are drawings of the 250-ft. double-track draw over the Calumet River in Chicago. This bridge was built by the Edge Moor Bridge Works, under the direction of W. L. Stebbings, Consulting Engineer. The drawings show clearly the principal features. There are 78 cast-steel wheels 14 in. diameter and 7 in. face. Four pinions with equalizing-gear similar to that used for many years by the Edge Moor Company are employed. The engine is a double reversible one, with cylinders $8\frac{1}{2} \times 12$ in. The method of distributing the load thoroughly over the drum is shown on Plates G and H. The most objectionable feature is the use of hollow wheels. The bridge sets low and the wheels during high storms are often under water and in winter are liable to be full of ice. The radial rods were found to be insufficient, and these, as well as the spacing-ring, were made heavier than shown and very much strengthened. Aside from these defects, which were easily remedied, the bridge is found to work in a very satisfactory manner and is giving excellent service. It carries a heavy traffic and is opened many times daily.

Plate K shows the engine-house of the Maumee River draw at Toledo, Ohio, giving the arrangement of boilers, coal-bin, water-tank, etc. Owing to the limited height required to clear bracing as finally designed, a flatter tin roof was used. This was an advantage as a greater protection against fire, but it did not have as attractive an appearance.

Air-motors.—The work done in an air-motor using cold air (temperature about 60°) and expanding adiabatically down to atmospheric pressure, allowing for the losses due to friction, etc., is given approximately by the following formula:

$$W = 72000\left[1 - \left(\frac{15}{p}\right)^{0.3}\right] \text{ ft.-lbs.,}$$

and if the air is heated before entering the motor

$$W = 72000\left[1 - \left(\frac{15}{p}\right)^{0.3}\right]\frac{T_1}{T}\text{ ft.-lbs.},$$

T and T_1 being the absolute temperatures before and after heating, or the temperature in Fahrenheit degrees $+ 461°$; p = the pressure at which the air enters the motor; W = the work done per pound of air.

Pipes and Cylinders, Strength and Dimensions of.—Let t = thickness of metal, D = inside diameter; then

$$\left.\begin{array}{l} \text{for water or gas } t = .32 \text{ in.} + \dfrac{D}{80}, \\ \text{" steam-pipes } t = .47 \text{ in.} + \dfrac{D}{50}, \end{array}\right\} \text{cast iron.}$$

Cast-iron steam-cylinder or pump-barrel: $t = .79 \text{ in.} + \dfrac{D}{100}$.

Hydraulic cylinder: P, the pressure per sq. in., $= \dfrac{S}{1 + \dfrac{r}{t}}$.

$$t = \frac{Pr}{S - P}.$$

r = the internal radius, t = the thickness.

Cylinder (Reuleaux): $P = S\left(\sqrt{1 + \dfrac{2t}{r}} - 1\right)$;

$$t = \frac{Pr}{S}\left(1 + \frac{P}{S}\right).$$

Hollow sphere: $P = \dfrac{2S}{1 + \dfrac{r}{t}}$; $t = \dfrac{Pr}{2(S - P)}$.

Flat circular plate supported at edges: $P = S\left(\dfrac{t}{r}\right)^2$.

" " " fixed " " $P = \dfrac{3}{2}S\left(\dfrac{t}{r}\right)^2$.

S = for cast iron 11000; S = for cast steel 36000; S = for wrought iron 20000; S = for wrought steel 25000.

HEAD OF WATER IN FEET AND THE EQUIVALENT PRESSURE IN POUNDS.

Feet Head.	Pounds per Square Inch.	Feet Head.	Pounds per Square Inch.	Feet Head.	Pounds per Square Inch.
1	.43	55	23.82	190	82.29
2	.87	60	25.99	200	86.62
3	1.30	65	28.15	225	97.45
4	1.73	70	30.32	250	108.27
5	2.17	75	32.48	275	119.10
6	2.60	80	34.65	300	129.93
7	3.03	85	36.81	325	140.75
8	3.40	90	38.98	350	151.58
9	3.90	95	41.14	375	162.41
10	4.33	100	43.31	400	173.24
15	6.50	110	47.64	500	216.55
20	8.66	120	51.97	600	259.85
25	10.83	130	56.30	700	303.16
30	12.99	140	60.63	800	346.47
35	15.16	150	64.96	900	389.78
40	17.32	160	69.29	1000	433.09
45	19.49	170	73.63		
50	21.65	180	77.96		

PRESSURE OF WATER IN POUNDS AND THE EQUIVALENT HEAD IN FEET.

Pounds per Square Inch.	Feet Head.	Pounds per Square Inch.	Feet Head.	Pounds per Square Inch.	Feet Head.
1	2.31	55	126.99	160	369.43
2	4.62	60	138.54	170	392.52
3	6.93	65	150.08	180	415.61
4	9.24	70	161.63	190	438.90
5	11.54	75	173.17	200	461.78
6	13.85	80	184.72	225	519.51
7	16.16	85	196.26	250	577.24
8	18.47	90	207.81	275	643.06
9	20.78	95	219.35	300	692.69
10	23.09	100	230.90	325	750.41
15	34.63	110	253.98	350	808.13
20	46.18	120	277.07	375	865.89
25	57.72	130	300.16	400	922.58
30	69.27	140	323.25	500	1154.48
35	80.81	150	346.34		
40	92.36				
45	103.90				
50	115.45				

The *theoretical horse-power* required to elevate water is found by multiplying the *gallons* pumped per *minute* by the total lift (including pipe-friction) *in feet* and dividing by 4000.

The *actual horse-power* for a 100-ft. lift is 1.7 times the theoretical horse-power, for a 200-ft. lift 1.45 times, and for a 300-ft. lift 1.25 times.

FRICTION OF WATER IN PIPES. (G. A. Ellis, C.E.)

FRICTION LOSS, IN POUNDS PRESSURE PER SQUARE INCH, FOR EACH 100 FEET OF LENGTH OF DIFFERENT SIZES OF CLEAN IRON PIPE DISCHARGING GIVEN QUANTITIES OF WATER PER MINUTE.

Gallons per Minute.	Sizes of Pipes—Inside Diameter.															Gallons per Minute.
	¾ in.	1 in.	1¼ in.	1½ in.	2 in.	2½ in.	3 in.	4 in.	6 in.	8 in.	10 in.	12 in.	14 in.	16 in.	18 in.	
5	3.3	0.84	0.31	0.12	0.05	......	......	......	......	......	......	......	......	......	......	5
10	13.0	3.16	1.05	0.47	0.12	......	......	......	......	......	......	......	......	......	......	10
15	28.7	6.98	2.38	0.97	0.30	0.11	......	......	......	......	......	......	......	......	......	15
20	50.4	12.3	4.07	1.66	0.42	0.15	......	......	......	......	......	......	......	......	......	20
25	78.0	19.0	6.40	2.62	0.51	0.21	0.10	......	......	......	......	......	......	......	......	25
30	......	27.5	9.15	3.75	0.91	0.33	0.11	......	......	......	......	......	......	......	......	30
35	......	37.0	12.4	5.05	1.20	0.45	0.17	......	......	......	......	......	......	......	......	35
40	......	48.0	16.1	6.52	1.60	0.52	0.22	......	......	......	......	......	......	......	......	40
45	......	......	20.2	8.15	2.00	0.65	0.28	......	......	......	......	......	......	......	......	45
50	......	......	24.9	10.0	2.44	0.81	0.35	0.09	......	......	......	......	......	......	......	50
75	......	......	56.1	22.4	5.32	1.80	0.74	0.17	......	......	......	......	......	......	......	75
100	......	......	......	39.0	9.46	3.20	1.31	0.33	0.05	......	......	......	......	......	......	100
125	......	......	......	......	14.9	4.89	1.99	0.53	......	......	......	......	......	......	......	125
150	......	......	......	......	21.2	7.00	2.85	0.69	0.10	......	......	......	......	......	......	150
175	......	......	......	......	28.1	9.46	3.85	1.00	......	......	......	......	......	......	......	175
200	......	......	......	......	37.5	12.47	5.02	1.22	0.17	......	......	......	......	......	......	200
250	......	......	......	......	......	19.66	7.76	1.89	0.26	0.07	0.03	0.01	......	......	......	250
300	......	......	......	......	......	28.06	11.2	2.66	0.37	0.09	0.04	......	......	......	......	300
350	......	......	......	......	......	......	15.2	3.65	0.50	0.12	0.05	0.02	......	......	......	350
400	......	......	......	......	......	......	19.5	4.73	0.65	0.16	0.06	......	......	......	......	400
450	......	......	......	......	......	......	25.0	6.01	0.81	0.20	0.07	0.03	......	......	......	450
500	......	......	......	......	......	......	30.8	7.43	0.96	0.25	0.09	0.04	0.017	0.009	0.005	500
750	......	......	......	......	......	......	......	......	2.21	0.53	0.18	0.08	......	......	......	750
1000	......	......	......	......	......	......	......	......	3.88	0.94	0.32	0.13	0.062	0.036	0.020	1000
1250	......	......	......	......	......	......	......	......	......	1.46	0.49	0.20	......	......	......	1250
1500	......	......	......	......	......	......	......	......	......	2.09	0.70	0.29	0.135	0.071	0.040	1500
1750	......	......	......	......	......	......	......	......	......	......	0.95	0.38	......	......	......	1750
2000	......	......	......	......	......	......	......	......	......	......	1.23	0.49	0.234	0.123	0.071	2000
2250	......	......	......	......	......	......	......	......	......	......	......	0.63	......	......	......	2250
2500	......	......	......	......	......	......	......	......	......	......	......	0.77	0.362	0.188	0.107	2500
3000	......	......	......	......	......	......	......	......	......	......	......	1.11	0.515	0.267	0.150	3000
3500	......	......	......	......	......	......	......	......	......	......	......	......	0.697	0.365	0.204	3500
4000	......	......	......	......	......	......	......	......	......	......	......	......	0.910	0.472	0.263	4000
4500	......	......	......	......	......	......	......	......	......	......	......	......	......	0.593	0.333	4500
5000	......	......	......	......	......	......	......	......	......	......	......	......	......	0.730	0.408	5000
Comp. disch'g power pipes.	$\sqrt{d^5}$	1	1.75	2.76	5.66	9.88	15.59	32	88.2	181	316.2	498.8	733.4	1024	1375	Comp. disch'g power pipes.

Springs.—Flat, spiral torsion-spring:

$$\text{Safe load } P = \frac{Sbh^2}{6R}; \quad \text{deflection} = \frac{12PlR^2}{Ebh^3}.$$

Flat, helical torsion-spring:

$$\text{Safe load } P = \frac{Sbh^2}{6R}; \quad \text{deflection} = \frac{12PlR^2}{Ebh^3}.$$

Round, helical torsion-spring:

$$\text{Safe load } P = \frac{S\pi d^3}{32R}; \quad \text{deflection} = \frac{64PlR^2}{\pi Ed^4}.$$

R = radius of circle at circumference of which P is applied.
d = diameter of spring-wire.
E = for tempered steel 42600000; S = 60000.
E = for ordinary steel 28400000; S = 24000.
E = for brass 9250000; S = 4600.

See also "Plate-girder Draw-spans."

Specifications for Springs.—All springs to be made of crucible steel. They shall contain not more than 1.10 per cent carbon and not less than .90 per cent; not more than .06 per cent of phosphorus, .50 per cent of manganese, or .045 per cent of sulphur. The pitch shall be uniform and the diameter of coil exact at all points. The diameter shall be such as to give the compression or torsion strength specified, and under the given quiescent loads the change in shape shall be within 5 per cent of the specified amounts.

Friction in Gearing.—The common formulæ for the per cent of work lost by friction in gearing are:

External gears, involute teeth: $F_G = \pi\phi\left(\frac{1}{N} + \frac{1}{n}\right)\frac{3}{4}C;$

Internal " " " $F_G = \pi\phi\left(\frac{1}{N} - \frac{1}{n}\right)\frac{3}{4}C;$

External " cycloidal " $F_G = \pi\phi\left(\frac{1}{N} + \frac{1}{n}\right)\frac{1}{2}C;$

Internal " " " $F_G = \pi\phi\left(\frac{1}{N} - \frac{1}{n}\right)\frac{1}{2}C.$

$C = 1.8P$; $F_G =$ the per cent of work lost by the friction; $\phi =$ the coefficient of friction; $N =$ the number of teeth in the larger gear; $n =$ the number of teeth in the smaller gear; $C =$ the arc of contact of one pair of teeth.

The amount of the friction will depend upon the shape of the teeth, their surface, and the general fit and alignment of the several parts. An approximate rule which will give results as nearly accurate as any for the conditions as found in drawbridges is as follows:

External gears, involute teeth: $F_G = \frac{3}{4}\left(\frac{1}{N} + \frac{1}{n}\right)P$;

Internal " " " $F_G = \frac{3}{4}\left(\frac{1}{N} - \frac{1}{n}\right)P$;

External " cycloidal " $F_G = \frac{1}{2}\left(\frac{1}{N} + \frac{1}{n}\right)P$;

Internal " " " $F_G = \frac{1}{2}\left(\frac{1}{N} - \frac{1}{n}\right)P$.

$P =$ the pitch N and n as above.

For new cast-iron gears ϕ may be taken as .2.

For cut-iron gears or cast-iron gears when worn to a bearing $\phi = .15$.

The coefficients for cast-steel gears may be taken as .18 and .13.

Where the teeth are of special shape determine graphically the length of the surface slid over by one pair of teeth while in action; divide this by the pitch and multiply by the coefficient of friction.

Pivots.—The safe load on a bronze disk turning between two hardened-steel disks may be taken as 2750 lbs. per square inch. When at rest this load may be increased to 5500 lbs. per square inch.

See also page 62 of "Plate-girder Draw-spans."

Friction of a Spherical Pivot.—The formula for the work lost in friction in a hemispherical bearing is

$$L = 13.16 P\phi r^2 N.$$

$N =$ the number of revolutions, $\phi =$ the coefficient of friction, $P =$ load, and $r =$ the radius. When the bearing-surface is not a full hemispherical, substitute for r the radius of the projection upon a plane normal to the axis of rotation of the limiting line of bearing.

The Focre Required to Bend Ropes over Pulleys.—An approximate formula for the force required to bend rope is for a bend of 180°:

$$F \text{ for hemp rope} = \frac{3.3 + .22P}{R};$$

$$F \text{ " wire "} = \frac{1.11 + .11P}{R};$$

$$F \text{ " " " with hemp centre} = \frac{1.22 + .22P}{R}.$$

$P =$ the load on the rope, $R =$ the radius of the pulley.

For leather belts $F = \frac{.3aP}{R}$ where $a =$ the area of the belt.

Friction of Screw-gearing.—The work W does in overcoming the frictional resistance of screw-gearing is

$$W = \phi NP\sqrt{40r^2 + p^2}.$$

$N =$ number of revolutions, $P =$ load, $p =$ the pitch, $\phi =$ coefficient of friction, $r =$ the radius of the pitch line.

The Friction of a Rope on a Pulley.—The formula for rope-friction is

$$F = T - T_1 = T_1(10^{2.7\phi N} - 1).$$

T and T_1 = the tensions on the driving and driven sides respectively, N = the number of revolutions, and ϕ = the coefficient of friction.

$T = T_1 10^{2.7\phi N}$ where T is the driving-force.

Safe Load on Bearings.—Lignum vitæ, 2800 lbs. per square inch.

Brass, 300 lbs. per square inch.

Bronze, 1000 lbs. per square inch.

The coefficient of friction of Babbitt metal-lined boxes is .033, and a good working load is 300 to 600 lbs. per square inch.

As stated elsewhere all important bearings should be babbitted or have brass or composition bushing which may be easily renewed when worn. Care to be exercised also in the location of bearings.

Wedge Friction-wheels.—Let P_r = the pressure applied radially, P_c = the circumferential force, ϕ = the wedge-angle; then

$$\frac{P_r = P_c\left(\sin\frac{\phi}{2} + f\cos\frac{\phi}{2}\right)}{f}.$$

The angle of the wedge is made 25° to 30°.

Roller-bearings.—Let P = the load, l = the length, r = the radius, S = the unit-stress, E = the modulus of elasticity.

$$S = .83\sqrt[3]{E}\sqrt[3]{\left(\frac{P}{lr}\right)^2};$$

$$P = \tfrac{4}{3}lr\sqrt{\frac{S^3}{E}}.$$

S = 12000 for cast iron; E for cast iron = 14200000.
S = 13500 for wrought iron; E for wrought iron = 28500000.

$S = 30000$ for hardened steel; E for hardened steel $= 42700000$.

Collar-brake.—In a collar-brake operated by a toggle-joint let $\phi =$ the angle which the links make with a normal to the axis of the wheel; then

$$P_a = \frac{P_c \tan \phi}{f}.$$

The angle ϕ is made as small as possible. $P_a =$ pressure in direction of axis of wheel, $P_c =$ circumferential force.

Friction-coupling.—In a cone coupling, let $P_cR =$ the moment tending to rotate the shaft, $\phi =$ the angle of taper, and $P_a =$ the power to be applied in the direction of the axis to hold the clutch in working contact; then

$$P_a = \frac{P_cR}{R}\left(\frac{\sin \phi}{f} + \cos \phi\right).$$

f for iron on iron $= 0.15$.
f for wood on iron $= 0.30$.
f for wood on wood $= 0.45$.
The angle ϕ should not be less than about $10°$.

Leather Belts, Strength of, and Safe Loads which They can Transmit—The breaking strength per inch of width for belts $\frac{3}{16}$ in. thick, well tanned oxhide, is:

In solid leather, 675 lbs. per inch of width.

At rivet-holes of splices, 365 lbs. per inch of width.

At lacing-holes, 215 lbs. per inch of width.

The safe tension is about 50 lbs. per inch of width.

Double belts have $1\frac{1}{2}$ times the strength of single belts.

The horse-power transmitted by single belts H. P. $= \frac{D \times R \times W}{2750}$, and for double belts H. P. $= \frac{D \times R \times W}{1833}$.
$D =$ diameter, $R =$ number of revlutions per minute, W $=$ width of belt in inches.

Arc of Contact	Ratio of Power Transmitted.
90°	2.21
112½	1.72
120	1.6
135	1.4
150	1.24
157½	1.17
180 to	1
270	1

In the above rule the arc of contact is assumed to be 180°. When the pulleys are of different diameters find the arc of contact for the smaller pulley and divide the result obtained by the formula by the constant given opposite the number of degrees most nearly corresponding with the arc of contact as determined.

Cord-friction.—The friction of ropes and leather belts on the surface of the pulley or sheaves over which they run depends upon the angle of contact or upon the length of rope or belt in contact with the surface of the pulley at one time. It depends also upon the centrifugal force, which in turn depends upon the velocity. The radius of the pulley or sheave has no effect on the amount of friction produced, except as it affects the velocity.

The coefficient to be used will depend upon the character of the surface of the pulley and the material in the belt: for leather belt or hemp rope on a smooth cast pulley say .18, and on wood pulley .16; for wire rope on cast iron .10, and on wood .20. If the surface of the pulley is grooved or V-shaped the pressure against the sides, and consequently the friction, is increased in proportion as the angle grows more acute. The action is similar to the wedge.

The friction is equal to the difference between the tension in the driving and in the driven rope. If T and t represent these tensions, then $F = T - t$.

$$F = t(2.718^{fa(1-z)} - 1).$$

In this formula $z = \frac{3wv^2}{8.05S}$, where w = the weight of one cubic inch of the material in the rope or belt, v = the velocity in feet per second of the rope, and S = stress per square inch in the rope.

For leather and hemp w = nearly .035 lb., and for wire rope $w = .315$.

Stress in Rope due to Bending.—In addition to the direct tension in the rope there is the stress produced by bending the rope around the pulley. The amount of this stress is given by the formula

$$S = 14220000\frac{d}{R}.$$

d = the diameter of *one wire* and R = the radius of the pulley; this should be added to the tension to obtain the total stress. The total stress should not exceed 56000 lbs. for steel rope $\begin{cases} 28000 \text{ tension} \\ 28000 \text{ bending} \end{cases}$

Stress in Rope due to Weight.—In a long rope the weight of the rope itself will be an important item. The length of rope, hanging vertical, which will produce a stress equal to the maximum allowed stress, S, is $L = .25S$. The force required to bend ropes over pulleys is

for hemp rope $\qquad F_b = \frac{.46d^2P}{R}$,

and for wire rope $\qquad F_b = 1.08 + .09\frac{P}{R}$.

P = the load on the rope, R = the radius of the drum or pulley, and d = the diameter of one wire.

For well-oiled, new rope it is claimed the force required for bending does not exceed .001 of the working load, where

the pulleys are large in proportion to the diameter of the rope. The diameter of the pulley should not be less than 1000 times the diameter of one wire in the rope.

Proportions for Pulleys.—Breadth of face, $B = 1\frac{1}{4}$ the width of belt.

Number of arms, $A = \frac{1}{2}\left(5 + \frac{R}{B}\right)$; $R =$ radius.

Width of arms, $h = \frac{1}{4}B + \frac{1}{10}\frac{R}{A}$.

Thickness of arms, $b = \frac{1}{2}h$.
Thickness of rim $= \frac{1}{8}h$.
Thickness of metal in hub, $\frac{3}{4}h$ to h.
Length of hub $= B$.
Taper in arms (width) $= \frac{1}{8}h$.
Taper in arms (thickness) $= \frac{1}{8}b$.

Plough-steel Rope.—Wire ropes of very high tensile strength, which are ordinarily called "plough-steel ropes," are made of a high grade of crucible steel, which, when put in the form of wire, will bear a strain of from 100 to 150 tons per square inch.

Where it is necessary to use very long or very heavy ropes, a reduction of the dead weight of ropes becomes a matter of serious consideration.

It is advisable to reduce all bends to a minimum, and to use somewhat larger drums or sheaves than are suitable for an ordinary crucible rope having a strength of 60 to 80 tons per square inch.

Transmission of Power by and Friction of Chains.—The frictional forces in chain machinery are: (1) the axle-friction; (2) the friction of the links in sliding over each other; (3) the friction of the chain upon the surface of the sheave or drum. Where smooth or grooved sheaves are used the amount of power transmitted depends upon this last friction. If sprocket-wheels are used the action is similar to that of spur-

Hemp Ropes.

TABLE OF MANILA ROPE.

Diameter.	Circumference.	Weight per Foot.	Breaking Load.	
in.	in.	lbs.	tons.	lbs.
.239	¾	.019	.25	560
.318	1	.033	.35	784
.477	1½	.074	.70	1568
.636	2	.132	1.20	2733
.795	2½	.206	1.91	4278
.955	3	.297	2.73	6115
1.11	3½	.404	3.81	8534
1.27	4	.528	5.16	11558
1.43	4½	.668	6.60	14784
1.59	5	.825	8.20	18368
1.75	5½	.998	9.80	21952
1.91	6	1.19	11.4	25536
2.07	6½	1.39	13.0	29120
2.23	7	1.62	14.6	32704
2.39	7½	1.86	16.2	36288
2.55	8	2.11	17.8	39872
2.86	9	2.67	21.0	47040
3.18	10	3.30	24.2	54208
3.50	11	3.99	27.4	61376
3.82	12	4.75	30.6	68544
4.14	13	5.58	33.8	75712
4.45	14	6.47	37.0	82880

HORSE-POWER OF "STEVEDORE" TRANSMISSION-ROPE AT VARIOUS SPEEDS, WITH AN ARC OF CONTACT NOT LESS THAN 170 DEGREES.

Diam. of Ropes.	Speed of the Rope in Feet per Minute.											Smallest Diam. of Pulleys in Inches.
	1500	2000	2500	3000	3500	4000	4500	5000	6000	7000	8000	
½	1.45	1.9	2.3	2 7	3.0	3.2	3.4	3.4	3.1	2.2	0	20
⅝	2.3	3.2	3.6	4.2	4.6	5.0	5.3	5.3	4.9	3.4	0	25
¾	3.3	4.3	5.2	5.8	6.7	7.2	7.7	7.7	7.1	4.9	0	30
⅞	4.5	5.9	7.0	8.2	9.1	9.8	10.8	10.7	9.3	6.9	0	36
1	5.8	7.7	9.2	10.7	11.9	12.8	13.6	13.7	12.5	8.8	0	42
1¼	9.2	12.1	14.3	16.8	18.6	20.0	21.2	21.4	19.5	13.8	0	54
1½	13.1	17.4	20.7	23.1	26.8	28.8	30.6	30.8	28.2	19.8	0	60
1¾	18.0	23.7	28.2	32.8	36.4	39 2	41.5	41.8	37.4	27.6	0	72
2	23.1	30.8	36.8	42.8	47.6	51.2	54.4	54.8	50.0	35.2	0	84

Power Transmitted by Ropes (actual operation).—A ½-in. wire rope running on 9½-in. pulleys at 95 revolutions per minute transmits 50 H. P.; and a 1-in. rope on 12-in. pulleys at speed of 100 ft. per second transmits 330 H. P.

STANDARD HOISTING-ROPE.

With 19 wires to the strand.

SWEDISH CHARCOAL-IRON.

Trade Number.	Price per Foot in Cents.	Diameter.	Circumference in Inches.	Weight per Foot in Pounds of Rope with Hemp centre.	Breaking Strain in Tons of 2000 Lbs.	Proper Working Load in Tons of 2000 Lbs.	Circumference of New Manila Rope of Equal Strength.	Min. Size of Drum or Sheave in Feet.
1	76	$2\frac{1}{4}$	$6\frac{3}{4}$	8.00	74	15	14	13
2	60	2	6	6.30	65	13	13	12
3	52	$1\frac{3}{4}$	$5\frac{1}{2}$	5.25	54	11	12	10
4	41	$1\frac{5}{8}$	5	4.10	44	9	11	$8\frac{1}{2}$
5	37	$1\frac{1}{2}$	$4\frac{3}{4}$	3.65	39	8	10	$7\frac{1}{2}$
$5\frac{1}{2}$	31	$1\frac{3}{8}$	$4\frac{3}{8}$	3.00	33	$6\frac{1}{2}$	$9\frac{1}{2}$	7
6	26	$1\frac{1}{4}$	4	2.50	27	$5\frac{1}{2}$	$8\frac{1}{2}$	$6\frac{1}{2}$
7	21	$1\frac{1}{8}$	$3\frac{1}{2}$	2.00	20	4	$7\frac{1}{2}$	6
8	16	1	$3\frac{1}{8}$	1.58	16	3	$6\frac{1}{2}$	$5\frac{1}{4}$
9	12	$\frac{7}{8}$	$2\frac{3}{4}$	1.20	11.50	$2\frac{1}{2}$	$5\frac{1}{2}$	$4\frac{1}{2}$
10	10	$\frac{3}{4}$	$2\frac{1}{4}$	0.88	8.64	$1\frac{3}{4}$	$4\frac{3}{4}$	4
$10\frac{1}{4}$	$7\frac{1}{2}$	$\frac{5}{8}$	2	0.60	5.13	$1\frac{1}{4}$	$3\frac{3}{4}$	$3\frac{1}{2}$
$10\frac{1}{2}$	$6\frac{1}{2}$	$\frac{9}{16}$	$1\frac{5}{8}$	0.48	4.27	$\frac{3}{4}$	$3\frac{1}{2}$	$2\frac{3}{4}$
$10\frac{3}{4}$	5	$\frac{1}{2}$	$1\frac{1}{2}$	0.39	3.48	$\frac{1}{2}$	3	$2\frac{1}{4}$
10a	$4\frac{3}{4}$	$\frac{7}{16}$	$1\frac{3}{8}$	0.29	3.00	$\frac{3}{8}$	$2\frac{3}{4}$	2
$10\frac{7}{8}$	$4\frac{1}{2}$	$\frac{3}{8}$	$1\frac{1}{4}$	0.23	2.50	$\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{1}{2}$

CAST STEEL.

1	92	$2\frac{1}{4}$	$6\frac{3}{4}$	8.00	155	31	....	$8\frac{1}{2}$
2	72	2	6	6.30	125	25	....	8
3	60	$1\frac{3}{4}$	$5\frac{1}{2}$	5.25	106	21	....	$7\frac{1}{4}$
4	48	$1\frac{5}{8}$	5	4.10	86	17	15	$6\frac{1}{4}$
5	43	$1\frac{1}{2}$	$4\frac{3}{4}$	3.65	77	15	14	$5\frac{3}{4}$
$5\frac{1}{2}$	36	$1\frac{3}{8}$	$4\frac{3}{8}$	3.00	63	12	13	$5\frac{1}{2}$
6	30	$1\frac{1}{4}$	4	2.50	52	10	12	5
7	25	$1\frac{1}{8}$	$3\frac{1}{2}$	2.00	42	8	11	$4\frac{1}{2}$
8	19	1	$3\frac{1}{8}$	1.58	33	6	$9\frac{1}{2}$	4
9	15	$\frac{7}{8}$	$2\frac{3}{4}$	1.20	25	5	$8\frac{1}{2}$	$3\frac{1}{2}$
10	$11\frac{1}{2}$	$\frac{3}{4}$	$2\frac{1}{4}$	0.88	18	$3\frac{1}{2}$	7	3
$10\frac{1}{4}$	9	$\frac{5}{8}$	2	0.60	12	$2\frac{1}{2}$	$5\frac{3}{4}$	$2\frac{1}{4}$
$10\frac{1}{2}$	8	$\frac{9}{16}$	$1\frac{5}{8}$	0.48	9	$1\frac{1}{2}$	5	$1\frac{3}{4}$
$10\frac{3}{4}$	7	$\frac{1}{2}$	$1\frac{1}{2}$	0 39	7	1	$4\frac{1}{2}$	$1\frac{1}{2}$
10a	$6\frac{1}{2}$	$\frac{7}{16}$	$1\frac{3}{8}$	0.29	$5\frac{1}{2}$	$\frac{3}{4}$	$3\frac{3}{4}$	$1\frac{1}{4}$
$10\frac{7}{8}$	6	$\frac{3}{8}$	$1\frac{1}{4}$	0.23	$4\frac{1}{2}$	$\frac{1}{2}$	$3\frac{1}{2}$	1

PLOUGH-STEEL ROPE.

With 19 wires to the strand.

Trade Number.	Price per Foot in Cents.	Diameter in Inches.	Weight per Foot in Pounds.	Breaking Strain in Tons of 2000 Lbs.	Proper Working Load.	Min. Size of Drum or Sheave in Feet.
1	130	2¼	8.00	240	46	9
2	104	2	6.30	189	37	8
3	87	1¾	5.25	157	31	7½
4	70	1⅝	4.10	123	25	6
5	60	1½	3.65	110	22	5½
5½	50	1⅜	3.00	90	18	5¼
6	41	1¼	2.50	75	15	5
7	34	1⅛	2.00	60	12	4½
8	28	1	1.58	47	9	4¼
9	22	⅞	1.20	37	7	3¾
10	16½	¾	0.88	27	5	3½
10¼	12½	⅝	0.60	18	3½	3
10½	10½	$\frac{9}{16}$	0.48	13	2	2½
10¾	9	½	0.39	10	1½	2
10a	8½	$\frac{7}{16}$	0.29	8	1¼	1½
10⅞	8	⅜	0.23	7	1	1

With 7 wires to the strand.

Trade Number.	Price per Foot in Cents.	Diameter in Inches.	Weight per Foot in Pounds.	Breaking Strain in Tons of 2000 Lbs.	Proper Working Load.	Min. Size of Drum or Sheave in Feet.
15	27	1	1.50	45	9	5½
16	21	⅞	1.12	33	6½	5
17	16	¾	0.92	25	4½	4
18	13	$\frac{11}{16}$	0.70	21	4	3½
19	11	⅝	0.57	16	3¾	3
20	8½	$\frac{9}{16}$	0.41	12	2	2¾
21	7	½	0.31	9	1½	2½
22	5	$\frac{7}{16}$	0.23	5	¾	2
23	4½	⅜	0.21	4	⅝	1½
24	3¾	$\frac{5}{16}$	0.16	3.2	½	1¼
25	3	$\frac{9}{32}$	0.125	2.5	$\frac{7}{16}$	1

STEEL FLAT ROPES.

Width and Thickness in Inches.	Weight per Foot in Pounds.	Strength in Pounds.	Width and Thickness in Inches.	Weight per Foot in Pounds.	Strength in Pounds.
⅜ × 2	1.19	35700	½ × 3	2.38	71400
⅜ × 2½	1.86	55800	½ × 3½	2.97	89000
⅜ × 3	2.00	60000	½ × 4	3.30	99000
⅜ × 3½	2.50	75000	½ × 4½	4.00	120000
⅜ × 4	2.86	85800	½ × 5	4.27	128000
⅜ × 4½	3.12	93600	½ × 5½	4.82	144600
⅜ × 5	3.40	100000	½ × 6	5.10	153000
⅜ × 5½	3.90	110000	½ × 7	5.90	177000

TABLE OF TRANSMISSION OF POWER BY WIRE ROPES.

Diameter of Wheel in Feet.	Number of Revolutions.	Trade Number of Rope.	Diameter of Rope.	Horse-power.	Diameter of Wheel in Feet.	Number of Revolutions.	Trade Number of Rope.	Diameter of Rope.	Horse-power.
3	80	23	$\frac{3}{8}$	3	7	140	20	$\frac{9}{16}$	35
3	100	23	$\frac{3}{8}$	$3\frac{1}{2}$	8	80	19	$\frac{5}{8}$	26
3	120	23	$\frac{3}{8}$	4	8	100	19	$\frac{5}{8}$	32
3	140	23	$\frac{3}{8}$	$4\frac{1}{2}$	8	120	19	$\frac{5}{8}$	39
4	80	23	$\frac{3}{8}$	4	8	140	19	$\frac{5}{8}$	45
4	100	23	$\frac{3}{8}$	5	9	80	20 19	$\frac{9}{16}$ $\frac{5}{8}$	47 48
4	120	23	$\frac{3}{8}$	6	9	100	20 19	$\frac{9}{16}$ $\frac{5}{8}$	58 60
4	140	23	$\frac{3}{8}$	7	9	120	20 19	$\frac{9}{16}$ $\frac{5}{8}$	69 73
5	80	22	$\frac{7}{16}$	9	9	140	20 19	$\frac{9}{16}$ $\frac{5}{8}$	82 84
5	100	22	$\frac{7}{16}$	11	10	80	19 18	$\frac{5}{8}$ $\frac{11}{16}$	64 68
5	120	22	$\frac{7}{16}$	13	10	100	19 18	$\frac{5}{8}$ $\frac{11}{16}$	80 85
5	140	22	$\frac{7}{16}$	15	10	120	19 18	$\frac{5}{8}$ $\frac{11}{16}$	96 102
6	80	21	$\frac{1}{2}$	14	10	140	19 18	$\frac{5}{8}$ $\frac{11}{16}$	112 119
6	100	21	$\frac{1}{2}$	17	12	80	18 17	$\frac{11}{16}$ $\frac{3}{4}$	93 99
6	120	21	$\frac{1}{2}$	20	12	100	18 17	$\frac{11}{16}$ $\frac{3}{4}$	116 124
6	140	21	$\frac{1}{2}$	23	12	120	18 17	$\frac{11}{16}$ $\frac{3}{4}$	140 149
7	80	20	$\frac{9}{16}$	20	12	120	16	$\frac{7}{8}$	173
7	100	20	$\frac{9}{16}$	25	14	80	8 7	1 $1\frac{1}{8}$	141 148
7	120	20	$\frac{9}{16}$	30	14	100	8 7	1 $1\frac{1}{8}$	176 185

gearing; and in this case the friction of Case 3, instead of being a benefit, is so much resistance which must be overcome before useful work can be performed. The friction of the links on the sides and ends of the pockets may amount to as much as 45 per cent of the applied force, and under the most favorable conditions would be 20 to 25 per cent. With flat links on smooth cast drum or round links on grooved pulley the following formulæ may be used: $T =$ the tension on the driving side, $t =$ tension on driven side, $n =$ the number of half-revolutions of the chain on the drum, $P =$ the power transferred, and $S =$ the allowed unit stress per square inch $= 5000$ to 8500 lbs. Then

$$t = \frac{T}{1.37n} P = T - t = T\left(1 - \frac{1}{1.37n}\right); \quad T = \frac{P}{1 - \dfrac{1}{1.37n}}.$$

Let H. P. = the horse-power transferred, $a =$ the area of the round from which the link is made, or $\frac{1}{2}$ the area of flat pin link, $v =$ the velocity in feet per minute; then

$$\text{H. P.} = \frac{avS\left(1 - \dfrac{1}{1.37n}\right)}{16500}.$$

The axle-friction and the friction of the links in sliding upon each other must be determined and the power required to overcome them must be deducted from the H. P. as given by the formula above.

For chains running on smooth sheaves the per cent of work lost in the link-friction is about 7 per cent, for grooved wheels 2.5 per cent, and for sprocket-wheels 1.25 per cent. It will be seen that this per cent for the friction of the links on each other decreases as the amount of power which can be transferred increases. The axle-friction is determined as in spur-gearing, and will not be discussed here.

TABLE OF WEIGHTS AND STRENGTH OF SHORT-LINK IRON CHAINS.

Dimensions of Iron.	Average Weight per Foot.	Breaking Strain.	Dimensions of Iron.	Average Weight per Foot.	Breaking Strain.
in.	lbs.	lbs.	in.	lbs.	lbs.
3/16	.42	1731	1	10.0	49280
1/4	.91	3069	1 1/16	11.3	52790
5/16	1.22	4794	1 1/8	12.5	59226
3/8	1.5	6922	1 3/16	14.0	65960
7/16	2.0	9408	1 1/4	15.5	73114
1/2	2.5	12320	1 3/8	18.5	88301
9/16	3.2	15590	1 1/2	22.0	105280
5/8	4.1	19219	1 5/8	25.5	123514
11/16	5.0	23274	1 3/4	29.5	143293
3/4	5.8	27687	1 7/8	33.5	164505
13/16	6.6	32307	2	38.0	187152
7/8	7.7	37632	2 1/4	48.5	224448
15/16	8.9	43277	2 1/2	60.0	277088

CRANE-CHAIN.

3/8	1.5	8960	7/8	7.7	51520
7/16	2.0	13440	15/16	8.9	58240
1/2	2.5	15680	1	10.0	62720
9/16	3.2	22400	1 1/8	12.5	82880
5/8	4.1	26880	1 1/4	15.5	100800
11/16	5.0	31360	1 3/8	18.5	120960
3/4	5.8	38080	1 1/2	22.0	143360
13/16	6.6	44800			

Proof = ½ breaking strain.
Safe working load = ¼ " "

STRENGTH OF MANILA AND HEMP ROPES.

It is a safe rule, up to 5 in. circumference, to multiply the square of the circumference by 8, and the product will be the number of net 100 lbs. required to break the rope.

POWER EQUIVALENTS.

Continental H.P.	English H.P.	Kgm.-metres per Second.	Foot-pounds per Second.	Watts.	1 Grm. 1° C. per Second.	1 Lb. 1° F. per Second.	Ergs per Second.
1	0.9863	75	542.5	736	117	0.722	736×10^7
1.0139	1	76	550	746	179	0.712	746×10^7
0.0133	0.0132	1	7.233	9.81	2.35	0.00937	9.81×10^7
0.00184	0.00182	0.138	1	1.356	0.326	0.0013	1.356×10^7
0.00136	0.00134	0.102	0.737	1	0.240	0.00095	10^7
0.00566	0.00559	0.425	3.07	4.17	1	0.004	4.17×10^7
1.42000	1.40400	106.7	772	1057	251.3	1	1047×10^7

Specifications for Steel for Castings—All steel castings should have an ultimate strength of not less than 62500 lbs. per square inch, an elastic limit of not less than 30000 lbs. per square inch, should elongate not less than 15 per cent in 8 in., and show a reduction at point of fracture of not less than 20 per cent. The amount of phosphorus contained should not exceed 0.08 per cent. All steel castings should be thoroughly annealed. Phosphor-bronze should be of the best quality for the purpose for which it is to be used. The castings should contain 88 per cent of copper and 12 per cent of phosphorized tin, the phosphorized tin to contain 5 per cent phosphorus. Upon each casting should be cast suitable test-pieces, which, in breaking, must show a good uniform metal, and when broken in testing-machine should show at least 40000 lbs. ultimate tensile strength.

Formula for Babbitt Metal: 50 parts tin; 1 part copper; 5 parts antimony.

Cement for Setting Castings.—Portland-cement concrete used for filling track-castings should be made of one part Portland cement, one part sand, and one part limestone screenings. The cement for setting the racks should be made of 80 per cent brimstone, 5 per cent asphaltum, and 15 per cent bituminous limestone (by weight). These quantities should be placed in a heating-vessel at the same time and heated up to a temperature of 300 degrees Fahrenheit for about 30 minutes. The temperature should be reduced to about 250 degrees in order to let the heavy foam settle, after which the material should be run into moulds for cooling off. This material should be remelted without foaming and run under the tracks in such manner as to insure the filling of all the voids. The brimstone used should be commercial, unmixed (seconds). The asphalt should contain 93 per cent of asphaltic bitumen.

All masonry bolts imbedded in above-mentioned cement filling should be coated with a composition made as follows:

To 45 per cent melted asphalt add 55 per cent pulverized green limestone which has previously been heated up to 400 degrees. This to be thoroughly mixed at a temperature of 350 degrees. Run this into moulds to cool off. Remelt this material and coat the bolts and the bottom of the base casting as well as the cement filling of the same. After the track is set all edges of the casting coming in contact with the cement filling under the turntable-tracks should also be given a coat of the same paint.

Rust-joint.

Quick-setting.	Slow-setting.
1 part sal ammoniac.	2 parts sal ammoniac.
2 parts flour of sulphur.	1 " flour of sulphur.
80 " iron-borings.	200 " iron-borings.

Mix to a stiff paste with water.

SPECIFICATIONS FOR WILLIS AVE. DRAW-SPAN, N. Y. CITY.

THOMAS C. CLARK, Consulting Engineer.

(310-FT. SPAN, 60-FT. ROADWAY.)

The swing-bridge guard will be composed of yellow-pine timber cribs filled with stone at each end, yellow-pine piles, yellow-pine caps, and oak sheeting.

The cribs will be built of square sawed yellow-pine timber with dovetailed ties, the timbers breaking joint and thoroughly drift-bolted together with 20-in. by $\frac{7}{8}$-in. drift-bolts.

The cribs will finally be filled with approved stone, of not over 6 cubic feet in size, up to the backing-log. The square timber and piles will be cased in on the outside with 4-in. white-oak plank for 14 ft. in depth, spiked with 10-in. spikes in every timber on alternate edges to longitudinal waling-strips of yellow pine. A ribbon-piece of 6 in. by 10 in. with chamfered edges will be screw-bolted along the top edge all around bridge-guard, screw-heads being countersunk on outside face.

Cribs will be sunk to solid bearings on sand, and mud must be dredged away.

Boiler-plates of $\frac{3}{8}$-in. iron, the full depth of the sheeting, and lapping the corners 3 ft. either way, will be securely fastened at the ends and shoulders. The fender-planks across the interval of the centre pier will be supported and braced by a square timber trussing, well screw-bolted together.

The cribs will be brought to elevation and surfaced and levelled off with stone ballast, the whole work being finished in a neat and workmanlike manner. The iron used for bolts or spikes to be capable of being bent over on itself within the diameter of the piece without breaking.

The piles shall be of straight, sound yellow pine, measuring not less than 10 in. in diameter at the smaller end, and shall be driven to refusal with a hammer weighing 4000 lbs. falling 10 ft., or its equivalent.

They shall be of such length that the tops, where cut off in place, will be 6 ft. above high water.

The timber shall consist of sound, straight-grained, Southern yellow pine, free from sap, wane, shake, and large, loose, or unsound knots.

The caps will be drifted to each pile-head with $\frac{7}{8}$-in. round iron drift-bolts driven in holes bored $\frac{13}{16}$ in. diameter. All the planking, wales, and braces shall be bolted to the piles with $\frac{3}{4}$-in. screw-bolts.

OPERATING MACHINERY AND ENGINE-HOUSE.

The operating machinery will be placed in the engine-house, which will be supported on rolled-steel joists.

Floor to be covered throughout with double flooring of narrow pine matched, except around boilers, where $\frac{1}{2}$-in. diamond-pattern cast plates will be used. All exposed wood-work on inside of boiler-room to be protected with zinc. Sides and roof of engine-room to be cased with yellow pine and varnished.

Partitions of similar material shall be placed where shown on plan.

All to be painted outside two coats white lead and oil, roof two coats red lead and oil, windows and doors three coats white lead and oil.

The engines and boilers will be in duplicate, so that in case of injury to one the other may be able to do the work, and will be constructed as follows:

The steam operating-plant to be located in engine-house overhead, and to consist of two 10-in. by 7-in. double cylinder inclined centre-crank oscillating engines, each separately coupled direct to a differential gear-machine in proportion of 19 to 1 of sufficient strength to carry the full power of the engines. The gear-machines to be fitted with fine hammered-steel fulcrum-pins, bronze rollers, and bronze bushings, and to operate directly by one set of spur-gears through a suitable-size friction-clutch on the horizontal turning-shaft, and which, at a speed of from zero to average of eight to nine revolutions per minute, shall have sufficient grip to transmit the power of the engine. The outer ends of the horizontal shafting connecting with the vertical turning-shafting to be through suitable-size bevel-gears.

The two engines, with their differential gear-machines, friction-clutches, and mechanism to be in duplicate design throughout, and arranged so that each engine can run its own side independent of that of the other, but both controlled in their movements of starting and stopping by one reverse-valve and starting-mechanism.

Both engines, however, to be arranged so that they can be readily coupled together, and that when the whole duty of turning is to be done by one engine it will be done through a set of compound gears, which will give the vertical shaft about one half the speed given by the two engines.

SPEED OF OPERATING.

The speed of outer circumference of draw shall average 3 ft. per second, so as to open as follows:

243 ft., at 3 ft. per second....	81	seconds.
Time of operating end-cylinders.......	15	"
Total time of opening..........	96	"

HYDRAULIC WORK.

Each end of the bridge shall have a complete set of mechanism for centring, raising, and supporting the ends of the same, and to be operated by two hydraulic cylinders through connecting-rods placed at each end of the bridge. Each pair of cylinders (four) to have not less than 175 tons capacity at 2000 lbs. per square inch; to be double-acting and copper lined, with suitable size and length of hydraulic pipes, valves, and fittings for connecting the end-rams to the accumulators and duplex pumps, two of which of ample size are to be placed in the engine-room with a receiving-tank of sufficient size for filling cylinders, and the operation to be governed by pressure-valves.

All shafting in engine-room, as also the vertical turning- and locking-shafting, is to be of the best quality open-hearth steel, 56000 to 60000 lbs. ultimate strength, and not less than $\frac{25}{100}$ extension in 8 in., turned for whole length, and free from seams or other imperfections.

Vertical shafting shall be not less than 6 in. diameter and main horizontal shafts not less than $5\frac{1}{2}$ in. diameter. All shafting shall be provided with the necessary bearings of ample length and stiffness to properly support the shafts from spring. The bevel-gears throughout to be of hard and tough steel and annealed. Both of the upright turning-shafts to be provided with suitable-size brake-wheels, fitted with steel

bands and the necessary levers and connections, and which are to be separately operated by steam-cylinders from the engine-room.

Engine-room to be provided with suitable indicator of neat design, with engraved dials, for recording the movements of the draw in turning and locking. A full set of engineer's tools is to be provided, such as hammers, chisels, files, wrenches, and vises, together with all the necessary oilers, oil-cans, and oil-tanks, and a whistle of suitable size.

BOILERS.

Two (2) sixty (60) horse-power boilers to be provided of return tubular type; length of shell 70 in. by 66 in. wide, containing 168 2 in. diameter main tubes 70 in. long, and 62 3 in. diameter leg-tubes 5 in. long; grate-area 53 in. by 61 in.; boiler to be fitted with back combustion-chamber lined on the back, roof, and sides with Albemarle. Smoke-box to be neatly fitted to boilers and with hinged doors for cleaning out.

Boilers to be of best open-hearth, homogeneous steel tested sheets, and made in best manner throughout, and thoroughly braced, to be safe for a working pressure of 100 lbs. per square inch and tested by hydraulic pressure to 150 lbs. per square inch. Two separate suitable-size smoke-stacks to be provided, fitted with dampers and with the necessary roof-drums and umbrellas; all fittings and mountings of boiler to be of approved make and with the necessary ash-pans, grates, and firing-tools; the general design of boilers to be submitted for approval before construction.

Boilers to be cased in asbestos covered with Russian iron.

Boiler-room to be floored with ½-in. close diamond-pattern cast plates.

Two suitable-size Korting double-tube injectors and one Worthington duplex feed-pump to be provided, with all the

necessary pipes and pipe-covering valves and connections, for feeding boilers separately or together.

Water-tank to be provided of 1500 gallons capacity, with the necessary valvings and fittings, man- and hand-hole plates.

Two sets of hand-turning gear to be provided, independent of the steam-turning gear, operated through suitable capstan-heads of approved form, and with not less than four levers to each head, worked from deck of bridge.

Two duplicate sets of spare bevel-gears for main turning-shaft, one duplicate set of each size of the locking bevel-gears, one duplicate set of hand-turning gears, two extra pinions, and four extra rack-sections to be provided and finished for use.

There shall also be included a reversible iron coal-hod of half-ton capacity, properly rigged with hoisting-gear and crane and worked from the engine, and provided with tackle to hoist coal from the draw-rest to a chute connected with coal-bins in boiler-room.

ELECTRIC LIGHTING.

The bridge will be lighted with incandescent electric lights from a dynamo and engine on the draw-span.

There shall be furnished one vertical high-speed automatic cut-off engine, which shall be capable of developing not less than 65 horse-power when running at 300 revolutions per minute, with $\frac{1}{4}$ cut-off and 80 lbs. initial steam-pressure. Engine shall be of quality in all respects equal to engines of General Electric or B. F. Sturtevant Company. Engine shall be furnished with shaft-governor, which shall maintain the speed constant to within 2 per cent at any lead; shall have cylinder 8 in. diameter by 9 in. stroke, or its equivalent.

Engine is to be of the best design, workmanship, and material throughout, and is to be furnished with throttle-valve, oiling devices for all moving parts, sight-feed cylinder-lubri-

cator, release-valves, cylinder-cocks, complete oil-guards and -pans, and all necessary fittings, fixtures, and tools.

Engine is to be furnished, if necessary, with two fly-wheels of suitable size and design for driving the dynamo at its proper speed.

Engine is to be furnished with a cast-iron sub-base having a suitable extension upon which to place and secure the dynamo.

The contractor shall furnish the proper appliances for connecting the dynamo to the engine; also all the necessary piping, valves, and covering for connecting the engine to the boilers, together with all exhaust and grip piping and fittings.

There shall be furnished one continuous-current, compound-wound, direct-connected, multipolar dynamo, having a capacity of not less than 50 kilowatts at 125 volts pressure.

This dynamo shall be automatic in its operation and maintain a constant pressure at the lamps under all variations of lead.

It shall be furnished with self-oiling and self-aligning bearings, coupling for connecting to engine, Carpenter enamel or similar rheostat for shunt field and adjustable shunt for series field.

The brushes shall be easily accessible and capable of being adjusted or completely raised from the commutator by a single lever or hand-wheel. The dynamo shall be placed on the opposite side of the engine from the shaft-governor, and is to be secured to a suitably designed extension of the engine sub-base. Commutator is to be of the best tempered copper, insulated with mica. The design and construction shall be of the best, and the dynamo must be capable of delivering its full-rated load for twelve consecutive hours without sparking at the brushes or excessive heating. The insulation between the winding and the frame shall be not less than one megohm.

The commercial efficiency shall be not less than 90 per cent.

The dynamo shall not require a speed of more than 300 revolutious per minute, and shall be able to deliver 200 amperes at 125 volts pressure when running at this speed.

Dynamo shall run without noise or vibration; quality in all respects equal to those of General Electric Company, Westinghouse Electric & Manufacturing Company, Crocker & Wheeler, or Walker Co.

SWITCHBOARD.

The contractor shall furnish one marble or slate switchboard, conveniently located near the dynamo, upon which shall be placed the following instruments:

1 Weston station voltmeter, illuminated dial.

1 Weston station ammeter, illuminated dial.

1 shunt-field regulator of dynamo.

1 ground-detector.

1 switch for connecting ground-detector to any circuit.

The necessary switches and cut-outs for the various circuits and the complete equipment of the switchboard.

All switches are to be made of the knife-blade pattern, and are to be mounted upon marble or slate bases, except where they are directly attached to the switchboard.

The wires from the dynamo to the switchboard, and the wiring for the lights in the engine- and boiler-rooms, are to be run on porcelain insulators in a neat and substantial manner.

All sockets, cut-outs, junction-boxes, switches, and other appliances of any nature shall be constructed entirely of water-proof and incombustible materials.

All circuits leading from switchboards are to be protected by I. T. E. magnetic circuit-breakers.

WIRING CIRCUITS.

The distribution of the incandescent lights shall be approximately:

Cluster-lamps, 42 clusters of 6 lamps each, equal to 252, of 32 candle-power.

Below viaduct: bracket-lamps, 36, 32 candle-power.

Engine-room: bracket-lamps, 12, 16 candle-power.

Through spans: bracket-lamps, 24, 32 candle-power.

Totals, 312 of 32 candle- and 12 of 16 candle-power, but engineer reserves right to increase or diminish number of lights within power specified.

The wiring is to be done as follows:

The lights on the draw are to be placed one 32 candle-power in a fixture. All wires larger than No. 14 B. and S. are to be stranded. No wire smaller than No. 14 B. and S. is to be used.

The whole wiring system is to be so proportioned that the variation in pressure between any two lamps, at full load, shall not exceed two volts.

All lamps are to be 110 volts, 3½ watts per candle, or less, and have an average life of not less than 600 hours.

The wiring will be on the two-wire system.

All wires are to be tinned and insulated with rubber and covered with lead, and are to be the best grade of either of the following makes, Habershaw or Grimshaw.

The wires are to be run in iron-armored interior conduits, one wire in a tube. Separate circuits run from the engine-room switchboard as follows: One to each approach, two to the draw, and one to the engine-rooms.

For lighting the approaches two cables are to be run, one for each side of the river, and an extra set laid ready for connection. These cables are to be run down to the pivot of the draw and connected to a special device that will admit of the draw being opened without disturbing the lights on the ap-

proaches. This special device shall have no rubbing contacts or other working parts liable to derangement.

The submarine cables are to be stranded copper conductors insulated with pure gutta-percha, bedded with hempen yarn, and protected with galvanized steel armor wires.

The size of the conductor in these cables shall be determined by the cable-makers, and the insulation shall be not less than 200 megohms per mile.

This cable is to be laid by the contractor in a trench to be dredged for it. The lights on the approaches are to be placed in groups of six on the fixtures, of 16 candle-power each, wires to be run in iron-armored conduits, of size to be determined by engineer, which shall be laid under the roadway, a pocket or junction-box being provided in the base of each post for making connections to each group of lights and for testing.

From the ends of the submarine cable the necessary underground cable shall be laid in conduits and iron pipes, as before described, to supply the various groups of lights.

All connections from cable to cable, or from cable to branches, or from branches to fixture-wires, shall be made by means of water-tight junction-boxes.

These boxes shall be connected to the iron-pipe conduits by water-tight threaded joints.

The insulation between the entire system, with lamps connected and grounded, shall be not less than one half a megohm, and the resistance between the poles of the system shall be not less than one half megohm.

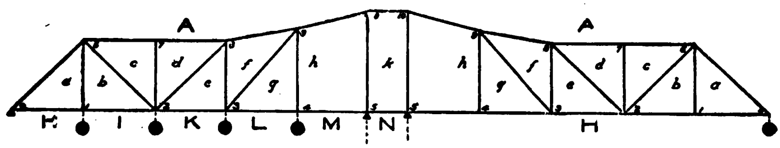


FIG. 59.

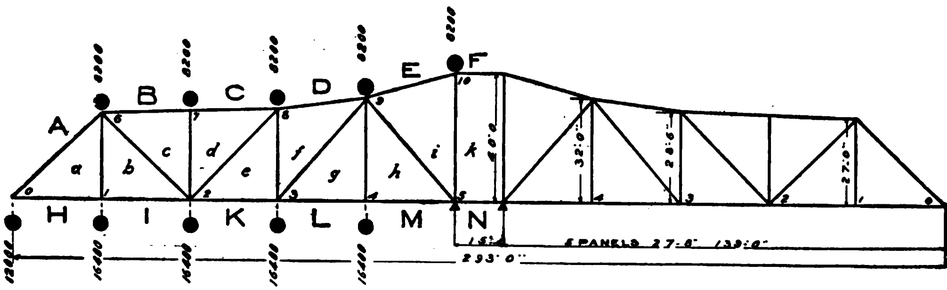


FIG. 60.
DEAD LOAD SWINGING.

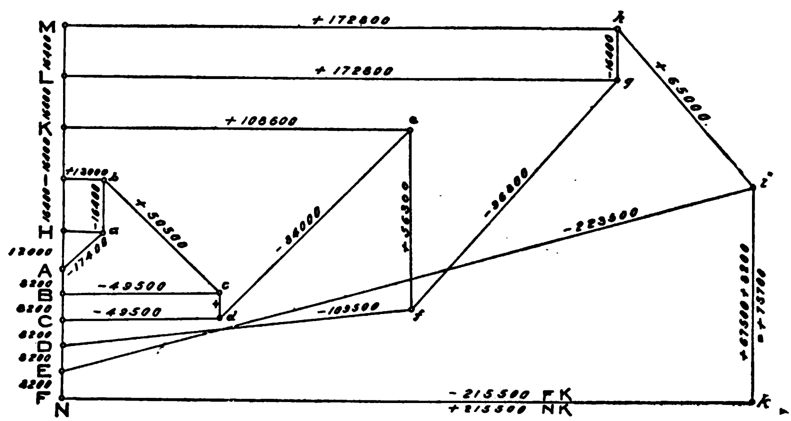


FIG. 61.
DEAD LOAD SWINGING.

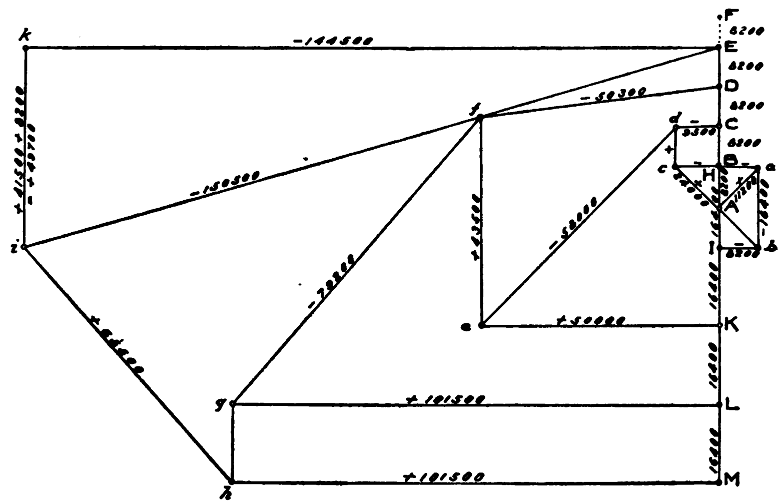


FIG. 62.

DEAD LOAD CONTINUOUS.

20000 upward reaction at end from wedge.

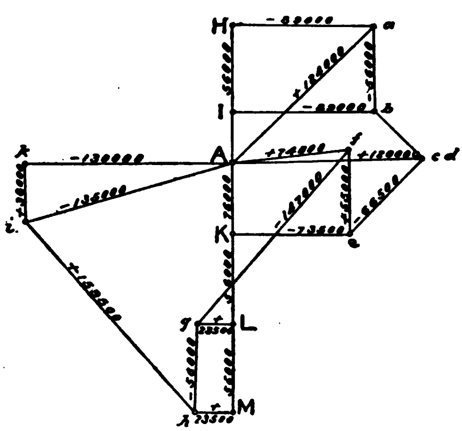


FIG. 63.

CONTINUOUS.

Live load on both arms, excess at 2 and 2.

Reaction at end = 105500 − 18480 = 87020 = *HA*.

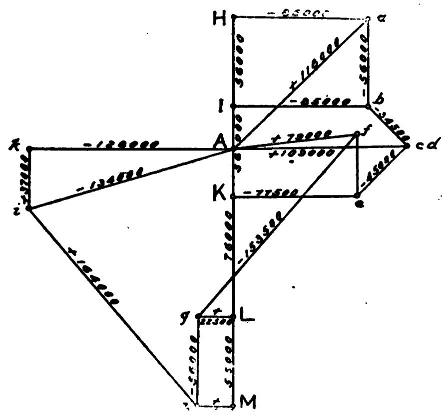


FIG. 64.

CONTINUOUS.

Live load on both arms, excess at 3 and 3.
Reaction at end = 101280 — 18720 = 82560 = HA.

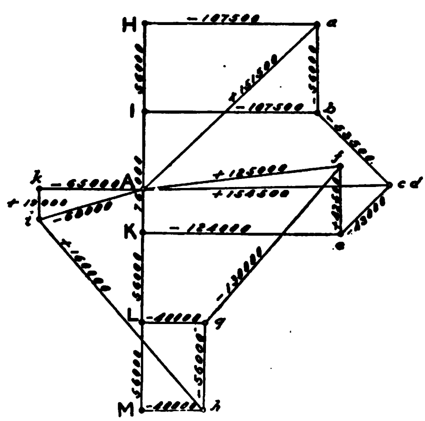


FIG. 65.

STRAINS IN LOADED ARM. CONTINUOUS.

Full live load on one arm, excess at 2. Load just coming on at far end of other arm. $R = + 105500$; $R_4 = - 18480$.

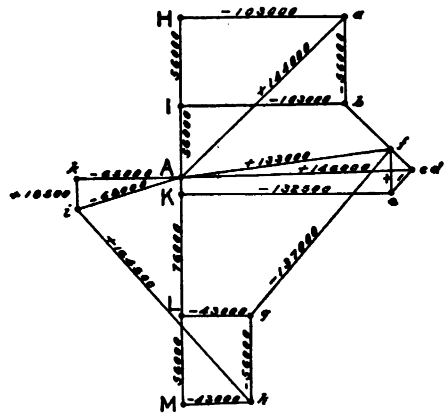


FIG. 66.

STRAINS IN LOADED ARM. CONTINUOUS.

Full live load on one arm, excess at 3. Advancing load at far end on other arm. $R_3 = + 101280$; $R_4 = - 18720$.

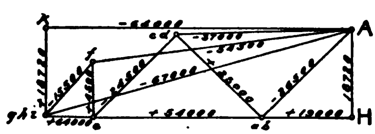


FIG. 67.

STRAINS IN UNLOADED ARM. CONTINUOUS.

Full load on one arm and advancing. Load at far end of other arm. $R = - 18720$.

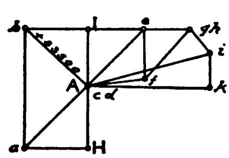


FIG. 68.

CONTINUOUS.

Full load on one arm and advancing. Load at HI (1) on other arm. Maximum compression in bc. $R = + 57150 - 18720 = + 38430$.

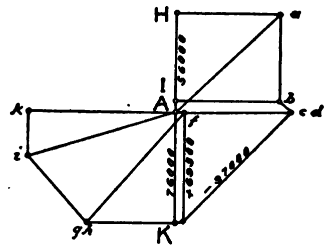


FIG. 69.
CONTINUOUS.

Full load on one arm and advancing. Load at *IK* (2) on other arm. Maximum stress in *de* and *ef*. $R = +81300 - 18720 = +62580$.

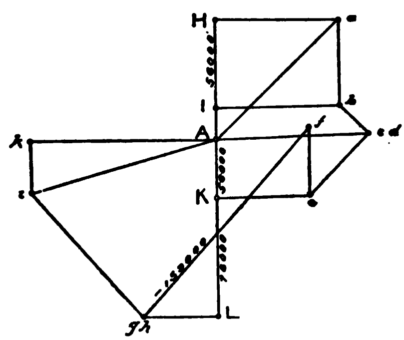


FIG. 70.
CONTINUOUS.

Full load on one arm and advancing load at *KL* (3) on other arm. Maximum stress in *fg*. $R = +94100 - 18720 = +75380$.

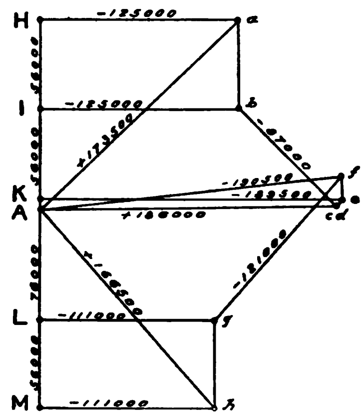


FIG. 71.

AS SINGLE SPAN.

Full load, excess at 3. Reaction $- AH = 120$.

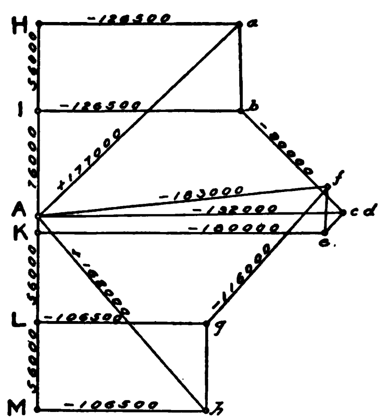


FIG. 72.

AS SINGLE SPAN.

Full load, excess at 2. Reaction $= AH = 124$.

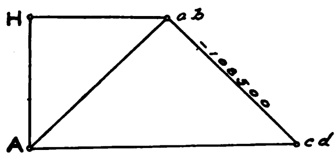


FIG. 73.
AS SINGLE SPAN.
Loads at 2, 3, and 4. Reaction — $AH = 79200$.

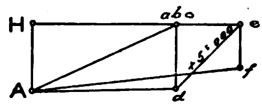


FIG. 74.
AS SINGLE SPAN.
Loads at 3 and 4. Reaction — $AH = 41600$.

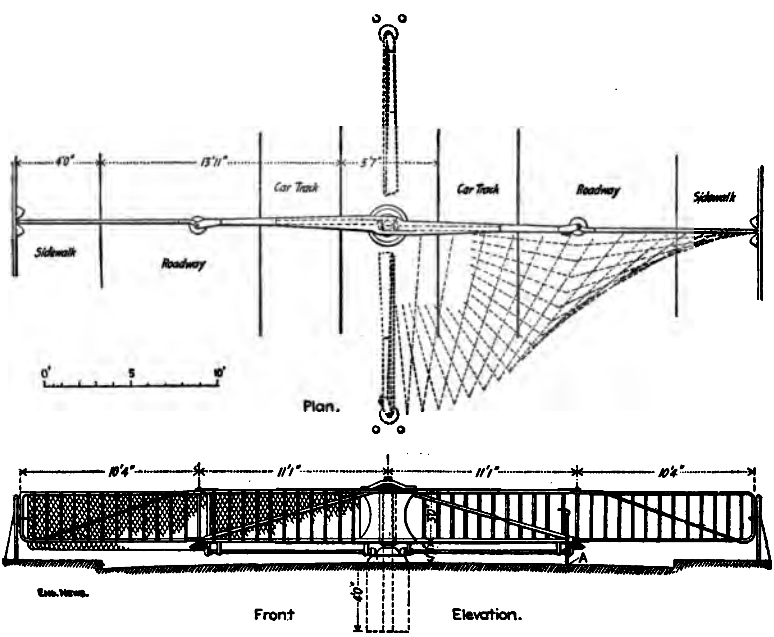


FIG. 75.
ELEVATION OF FOLDING GATE AT DRAWBRIDGE.

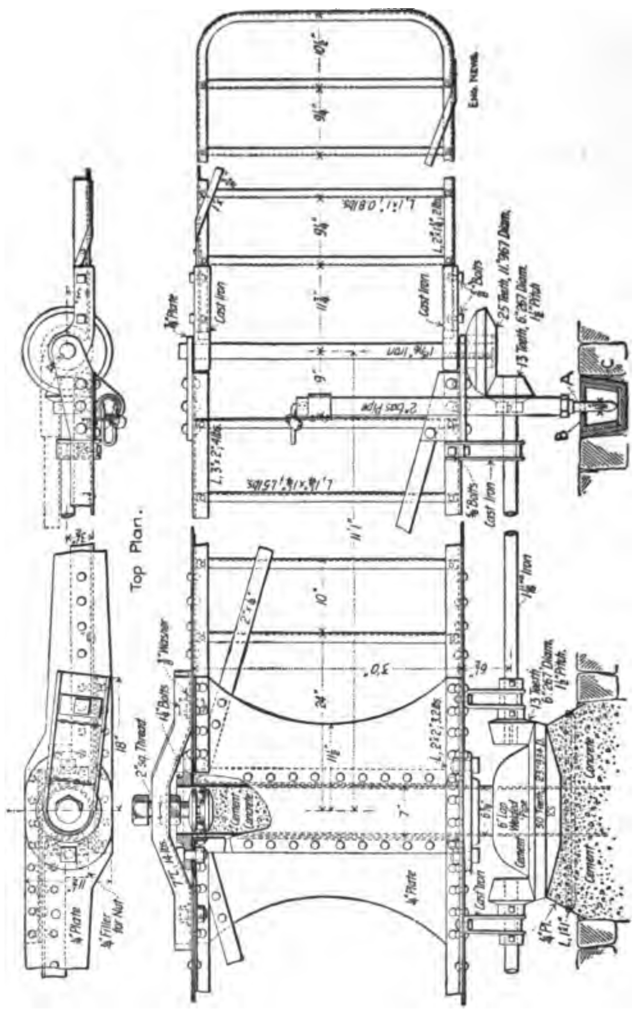


FIG. 76.
DETAILS OF FOLDING GATE AT DRAWBRIDGE.

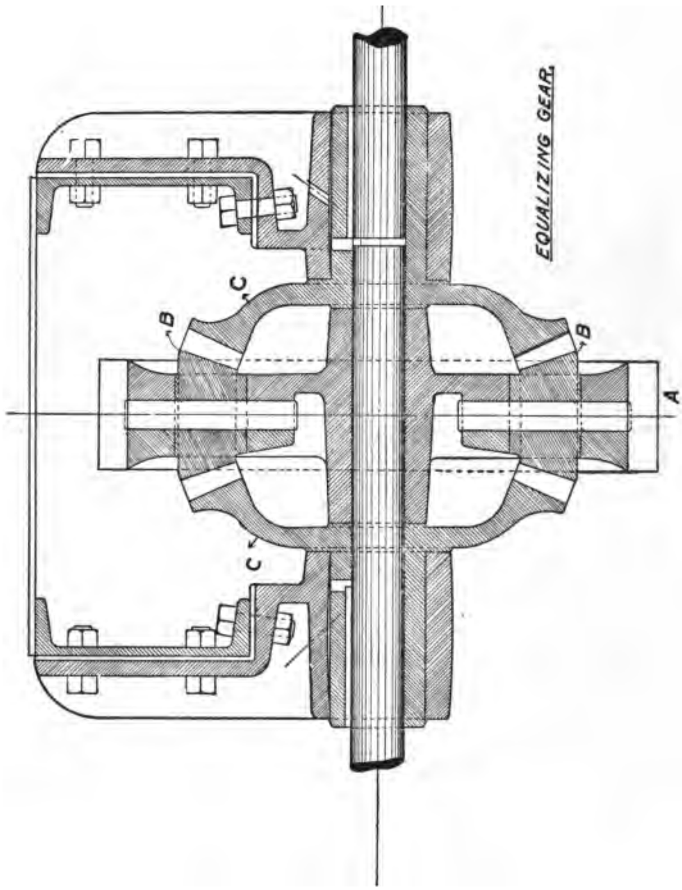


FIG. 77.
FOR DISTRIBUTING PRESSURE EQUALLY AT THE TWO PINIONS OF A DRAWBRIDGE.

LOCKING MACHINERY FOR DOUBLE-SWING BRIDGE, CLEVELAND, O.

WALTER P. RICE, M. AM. SOC. C. E., CHIEF ENGINEER.

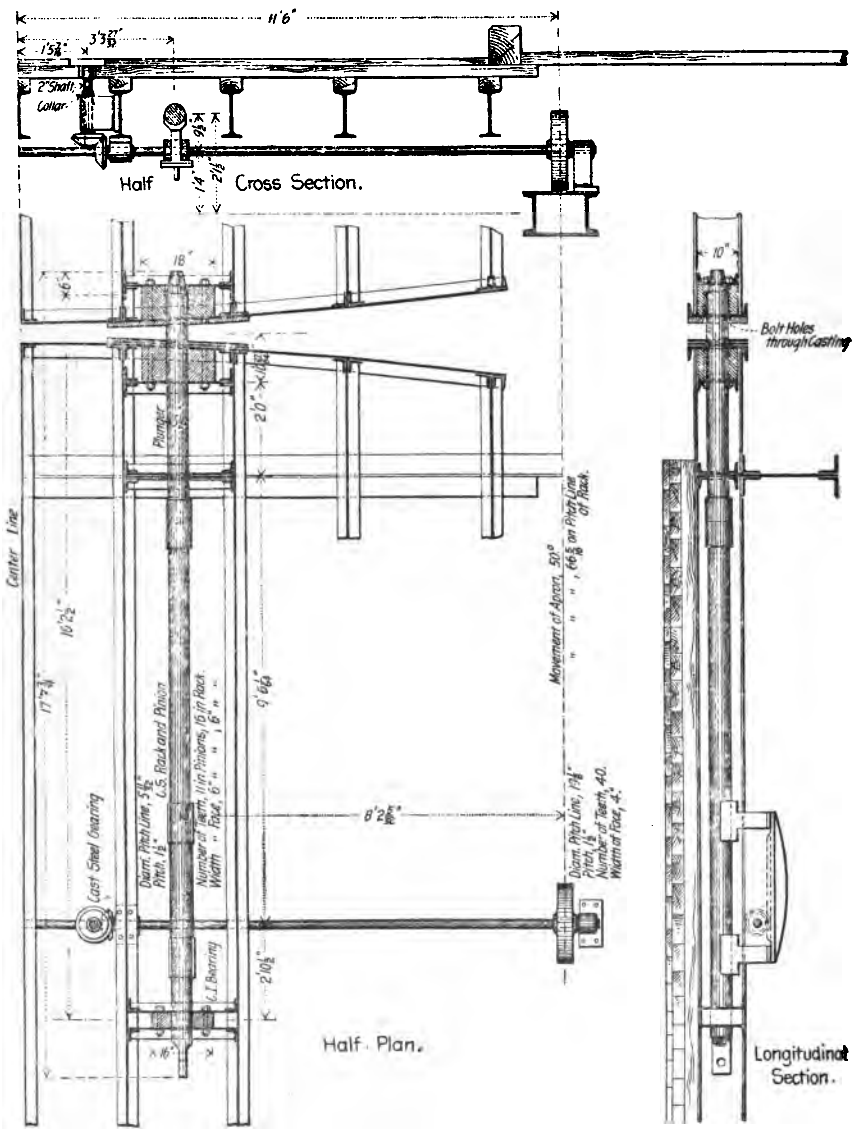


FIG. 79.
DETAILS OF CENTRE LOCKING DEVICE.

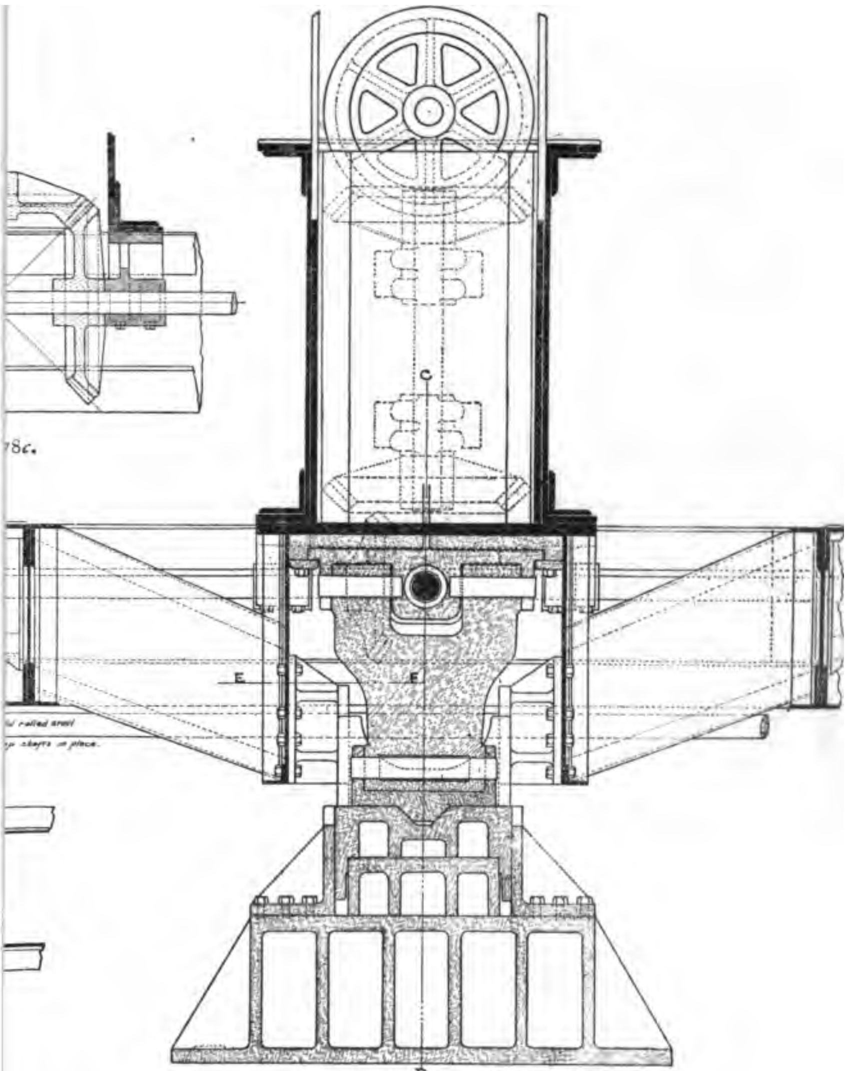


FIG. 78a.

LIFTING DEVICE UNDER END OF CENTER TRUSS
FOR
HARLEM RIVER FOUR TRACK DRAW
N.Y.C.&H.R.R.R.

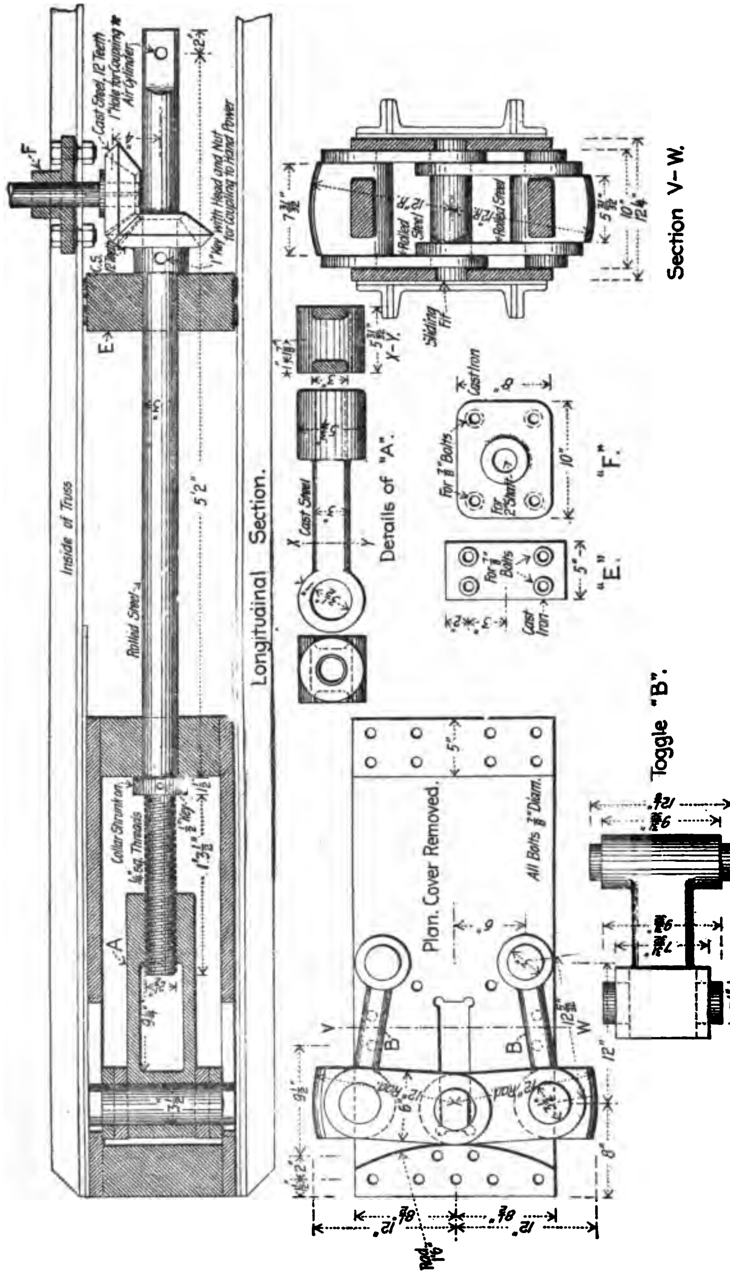


FIG. 79*a*.—DETAILS OF MACHINERY OF END-LOCK.

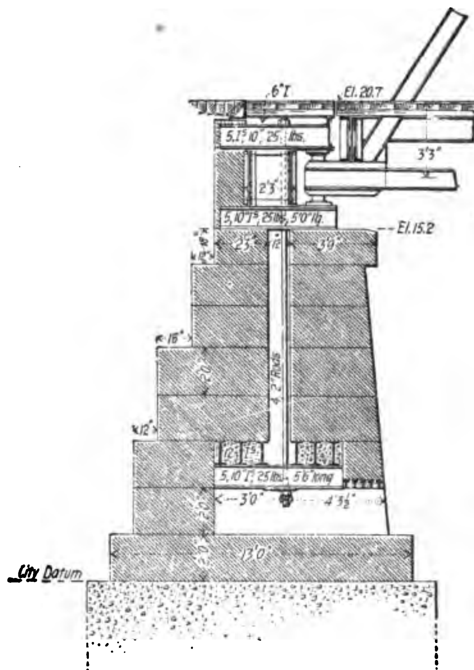


FIG. 80.
SECTION OF ABUTMENT SHOWING CONSTRUCTION OF ANCHORAGE.
SWING-BRIDGE AT CLEVELAND, OHIO.

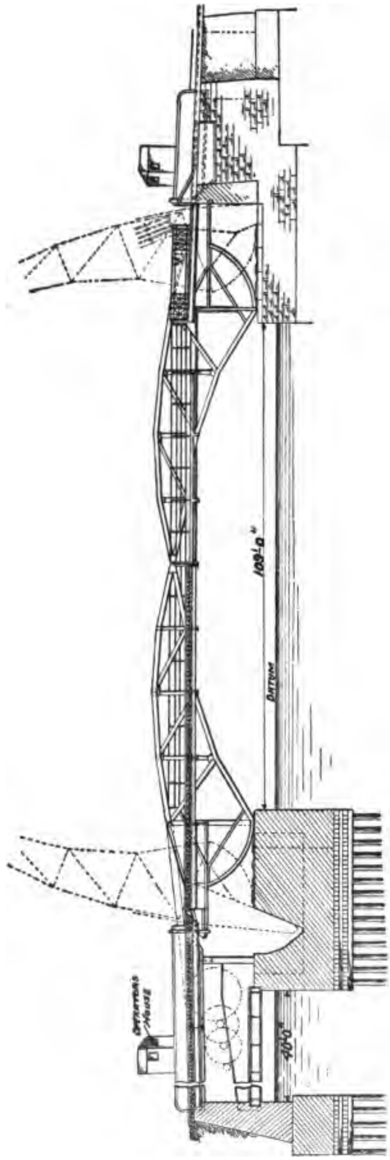


FIG. 81.
THE VAN BUREN STREET ROLLING LIFT-BRIDGE OVER THE CHICAGO RIVER, CHICAGO, ILL.

VAN BUREN STREET DRAW, CHICAGO.

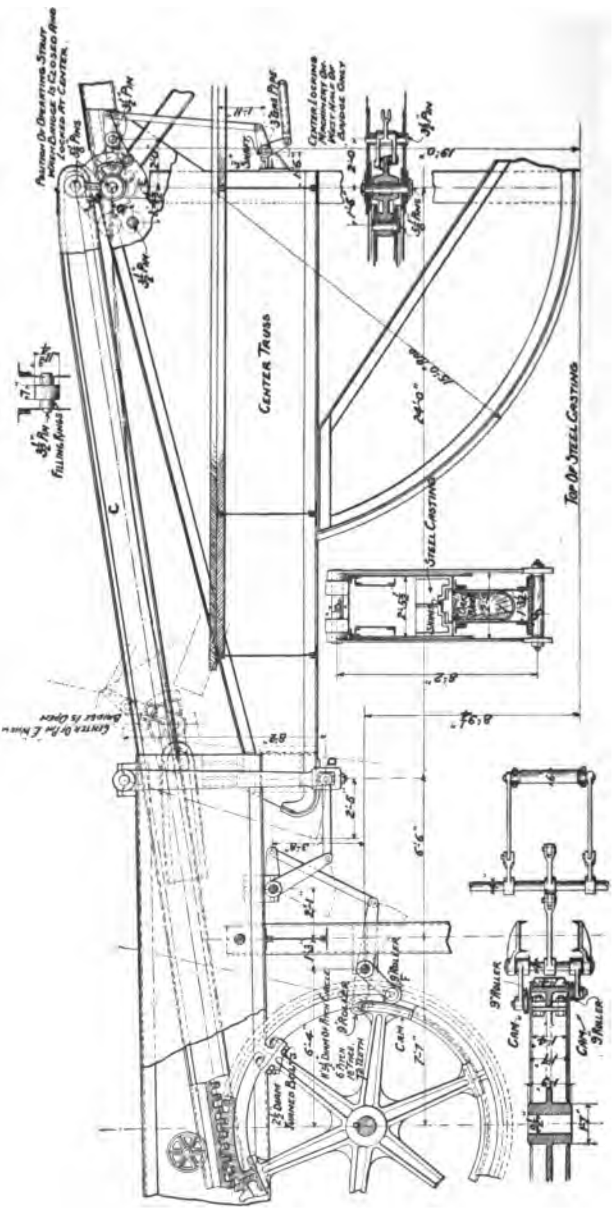


FIG. 82.
MACHINERY FOR APPLYING THE POWER FOR RAISING THE BRIDGE AND OPERATING THE LOCKING DEVICE.

VAN BUREN STREET DRAW, CHICAGO.

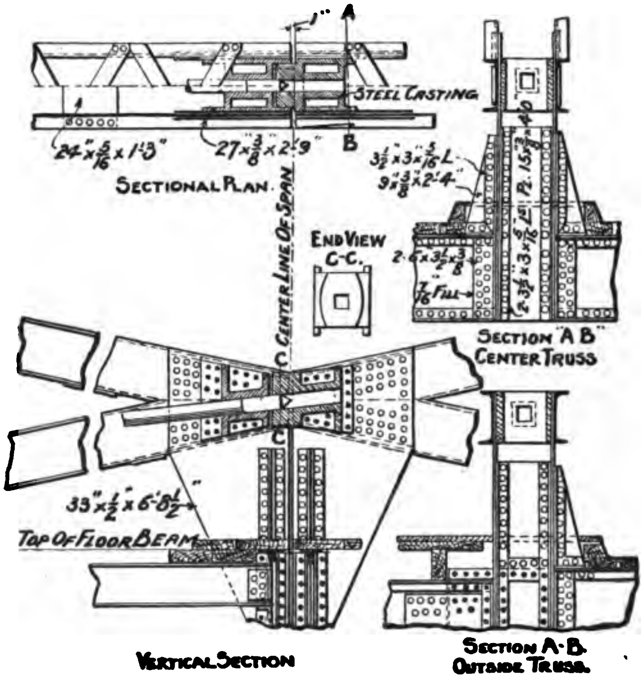


FIG. 82*a*.

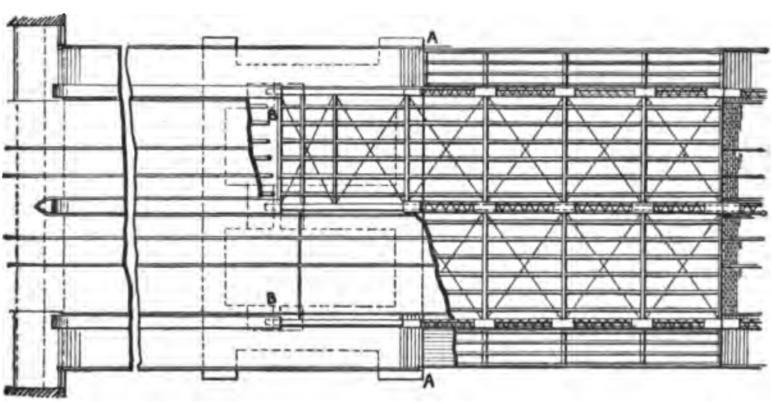
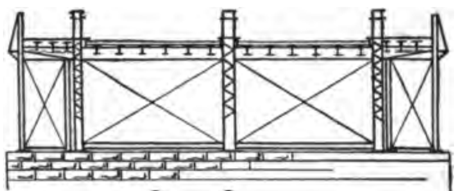


FIG. 82*b*.



CROSS SECTION.

FIG. 82*c*.

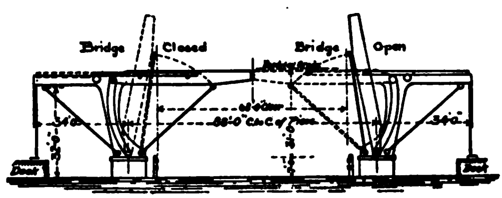


FIG. 83.
GENERAL DIAGRAM.

FOLDING SPAN, SIXTEENTH STREET VIADUCT, MILWAUKEE, WIS.

FOLDING SPAN, SIXTEENTH STREET VIADUCT, MILWAUKEE, WIS.

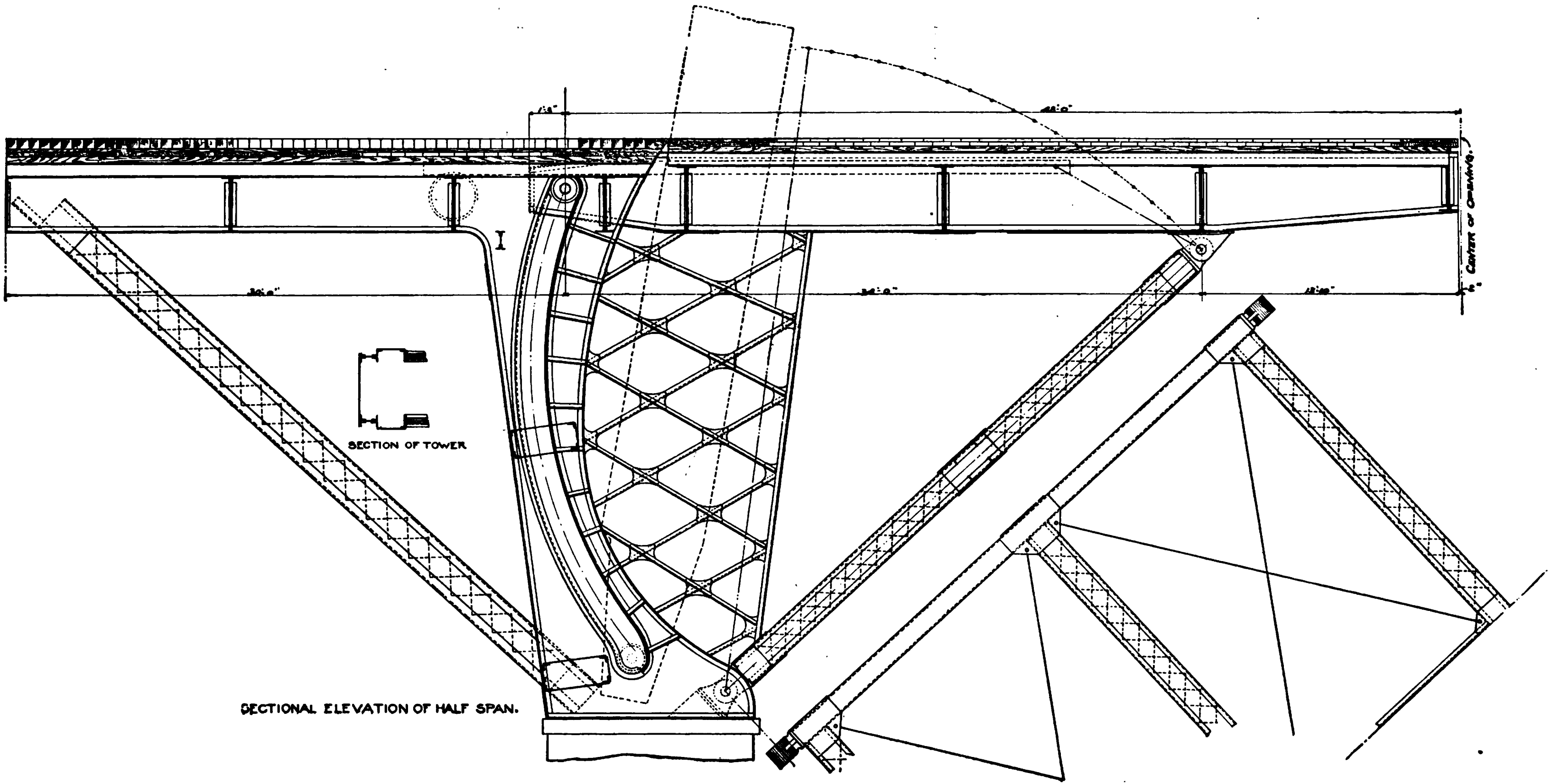


FIG. 84.—SECTIONAL ELEVATION OF HALF-SPAN.

SIXTEENTH STREET VIADUCT, MILWAUKEE, WIS.

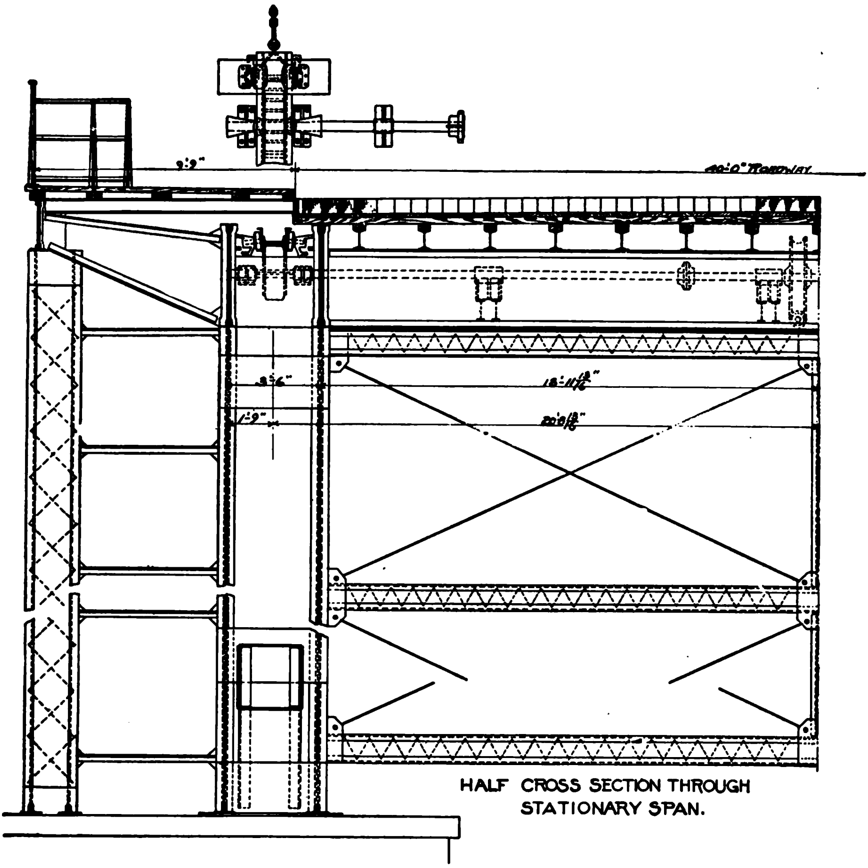


FIG. 84*a*.

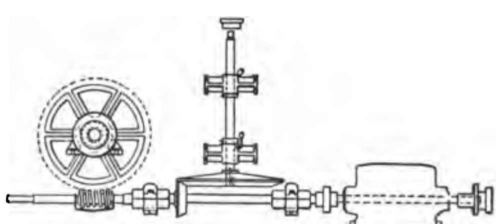


FIG. 84*b*.
SIDE ELEVATION OF MOTOR AND HAND TURNING GEAR.

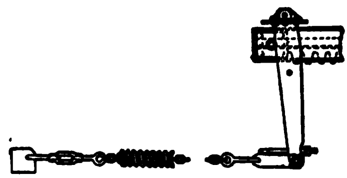


FIG. 84*c*.

CUSHIONED CONNECTIONS BETWEEN OPERATING-STRUT AND MOVING GIRDER.

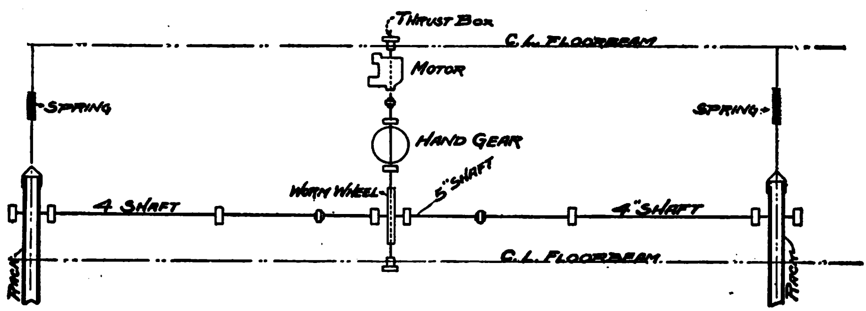


FIG. 84*d*.

DIAGRAM OF OPERATING MACHINERY.

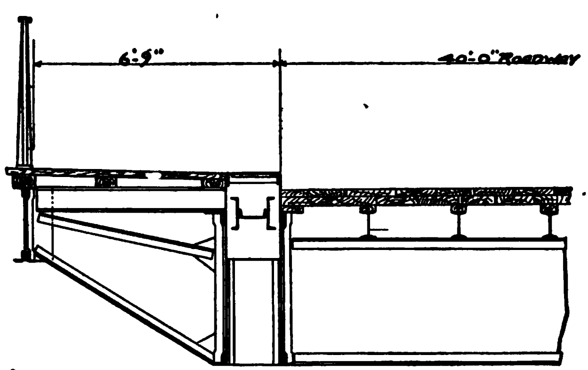
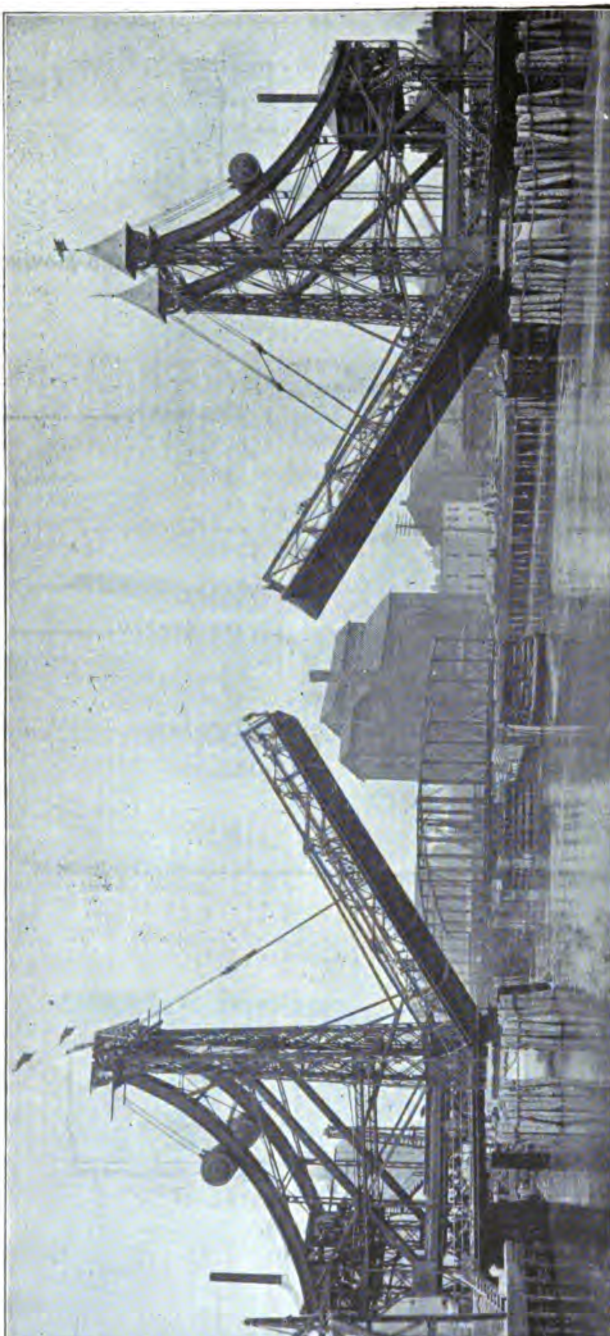


FIG. 84*e*.

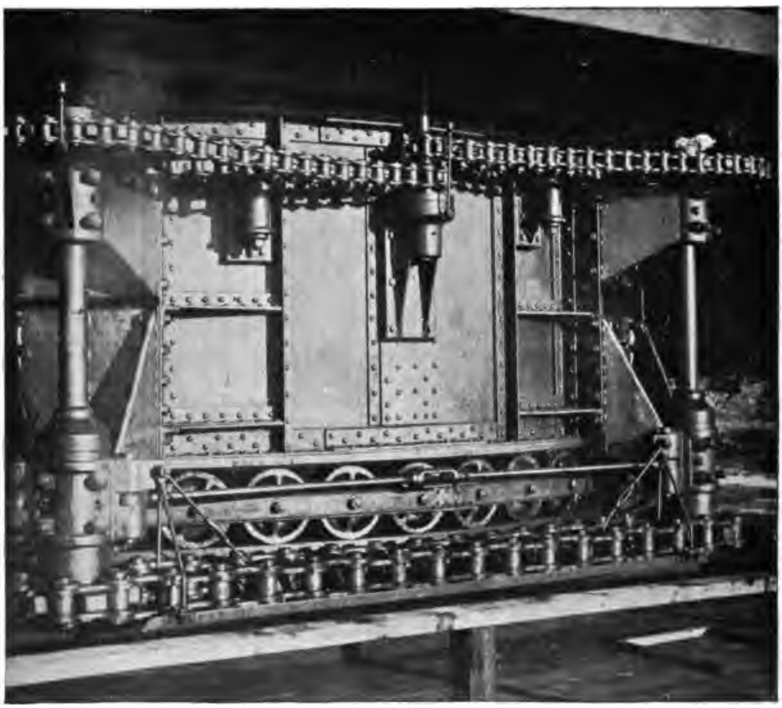
CROSS SECTION.



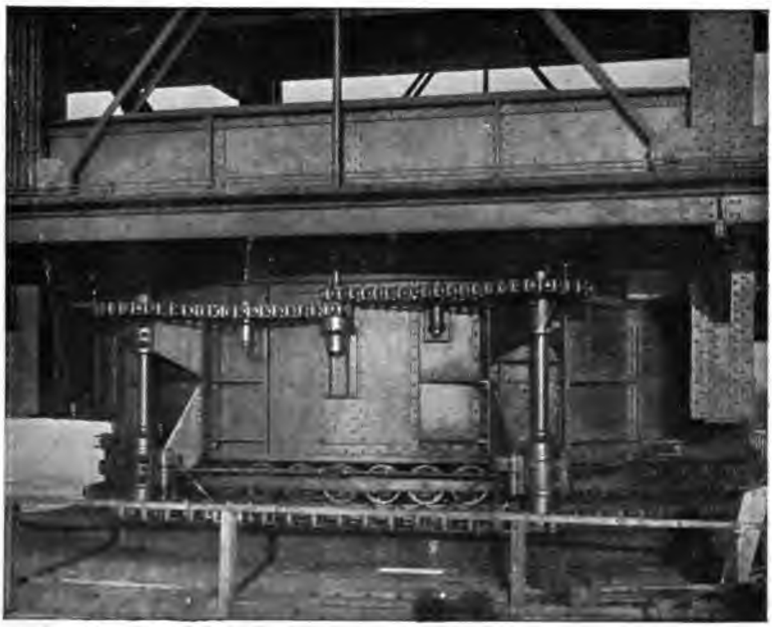
THE MICHIGAN AVENUE BASCULE BRIDGE, BUFFALO, N. Y.

Mr. E. B. Guthrie, M. Am. Soc. C. E., Chief Engineer Department of Public Works, Buffalo, and B. Weinhagen, Manager Wisconsin Bridge & Iron Company, Milwaukee, Wis., Engineers.

ROCK ISLAND BRIDGE

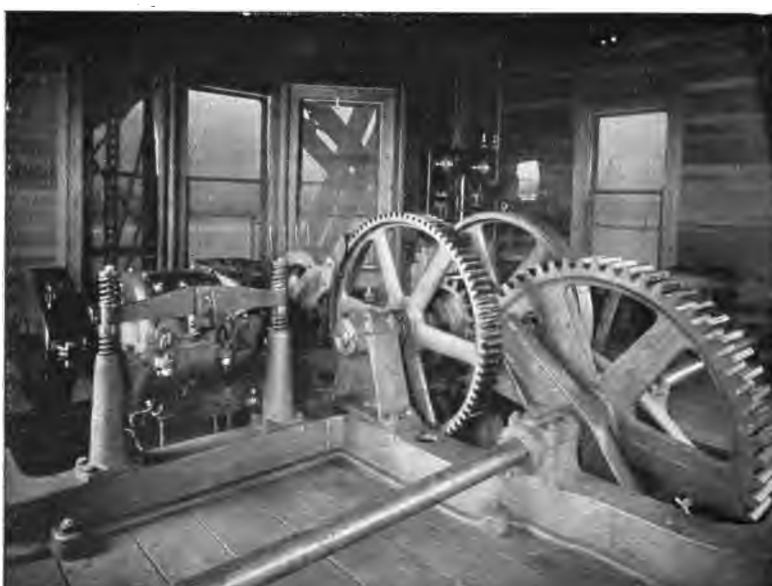


ROCK ISLAND BRIDGE.

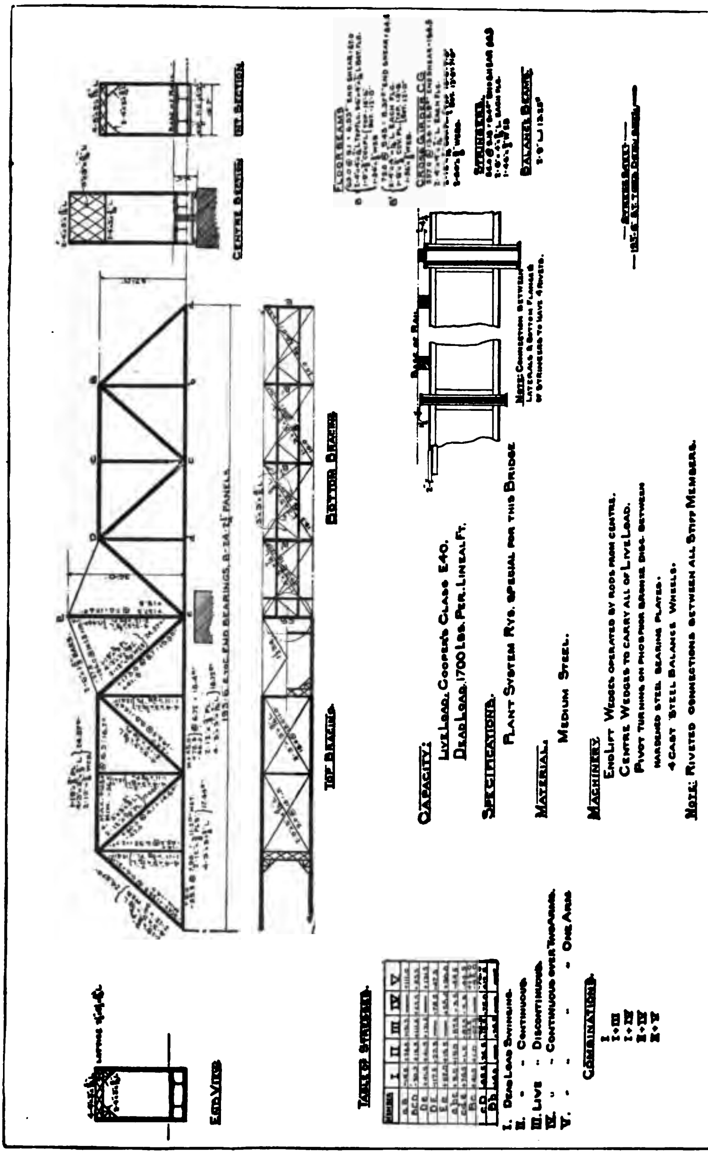


ROCK ISLAND BRIDGE.

ROCK ISLAND BRIDGE.



ROCK ISLAND BRIDGE.



STRESS SHEET FOR 193 FT. 6 IN. DRAW-SPAN FOR THE PLANT SYSTEM OF RAILWAYS.

INDEX.

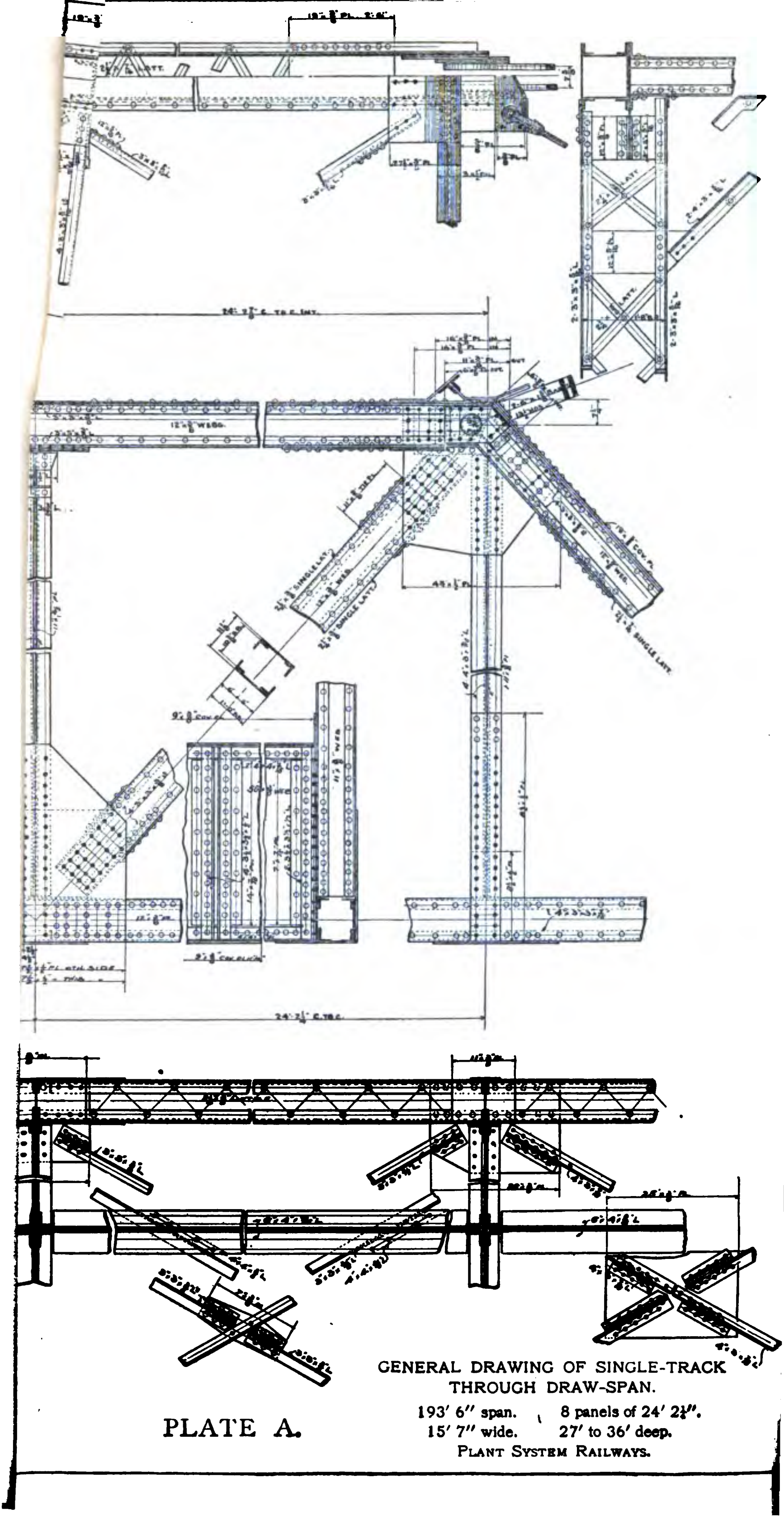


PLATE A.

GENERAL DRAWING OF SINGLE-TRACK THROUGH DRAW-SPAN.

193′ 6″ span. 8 panels of 24′ 2¼″.
15′ 7″ wide. 27′ to 36′ deep.

Plant System Railways.

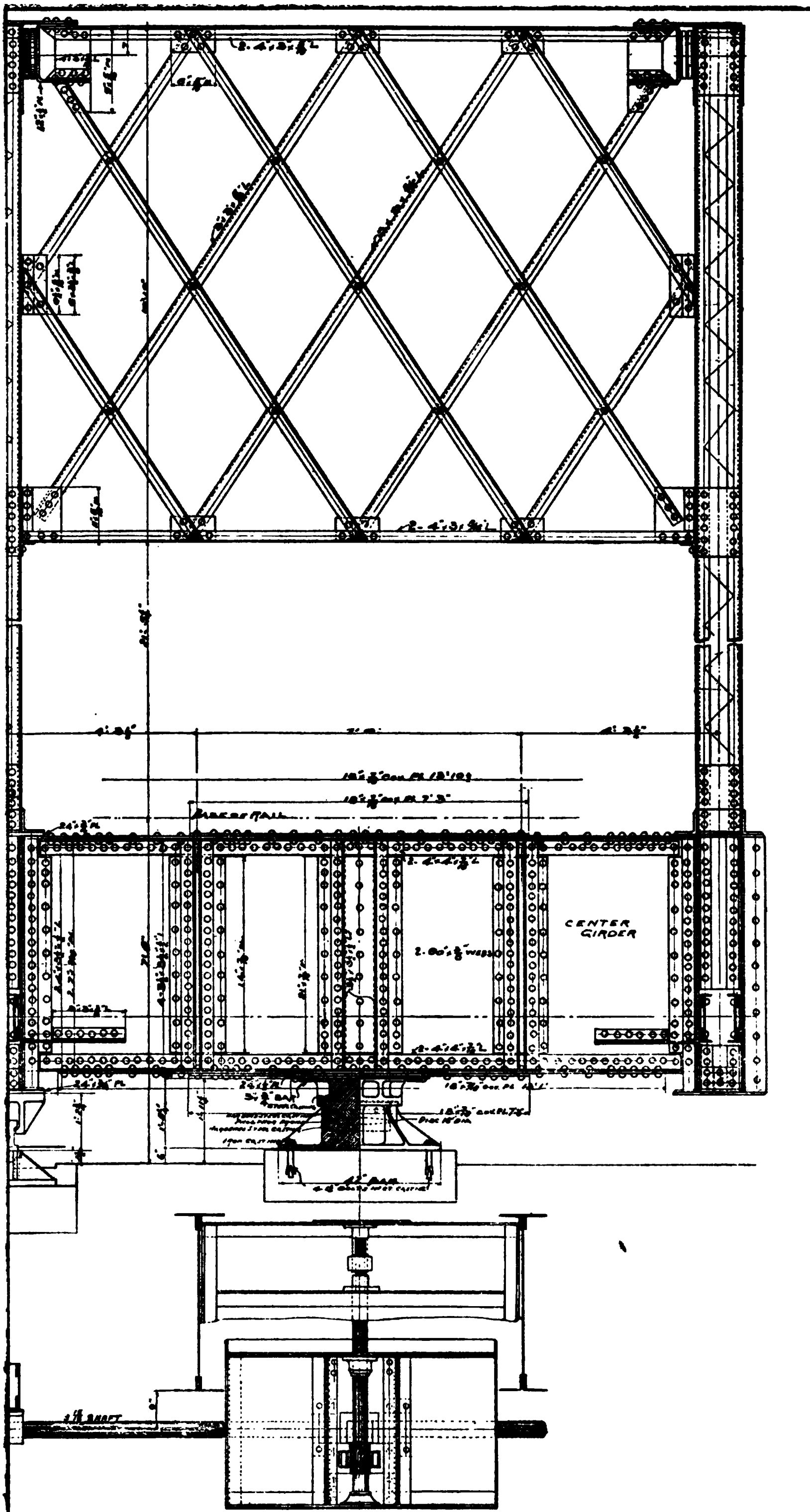


PLATE B.

GENERAL DRAWING OF SINGLE-TRACK
THROUGH DRAW-SPAN.

195′ 6″ span. 8 panels of 24′ 2½″.
15′ 7″ wide c. to c. 27′ and 36′ deep c. to c.

PLANT SYSTEM RAILWAYS.

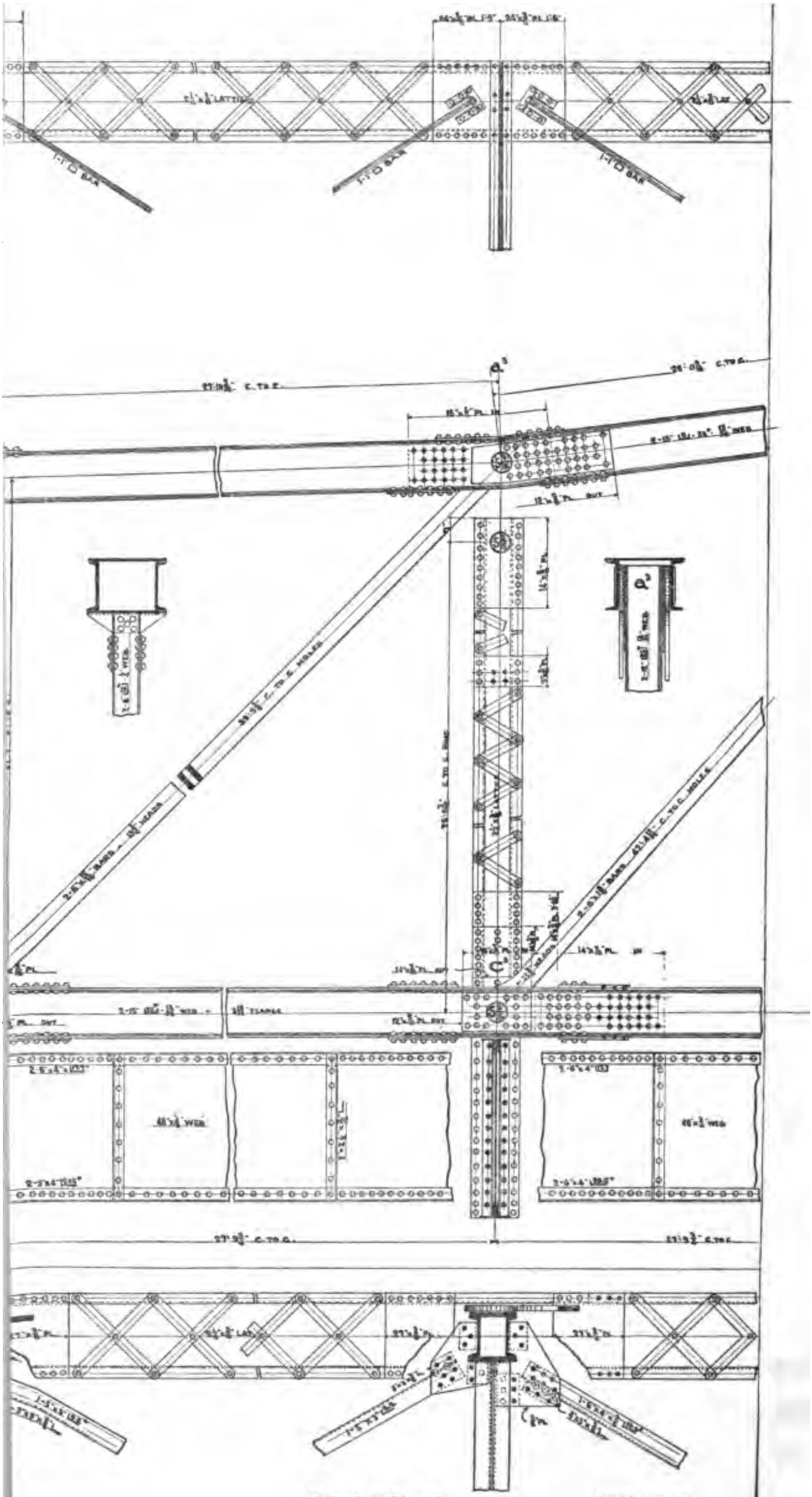
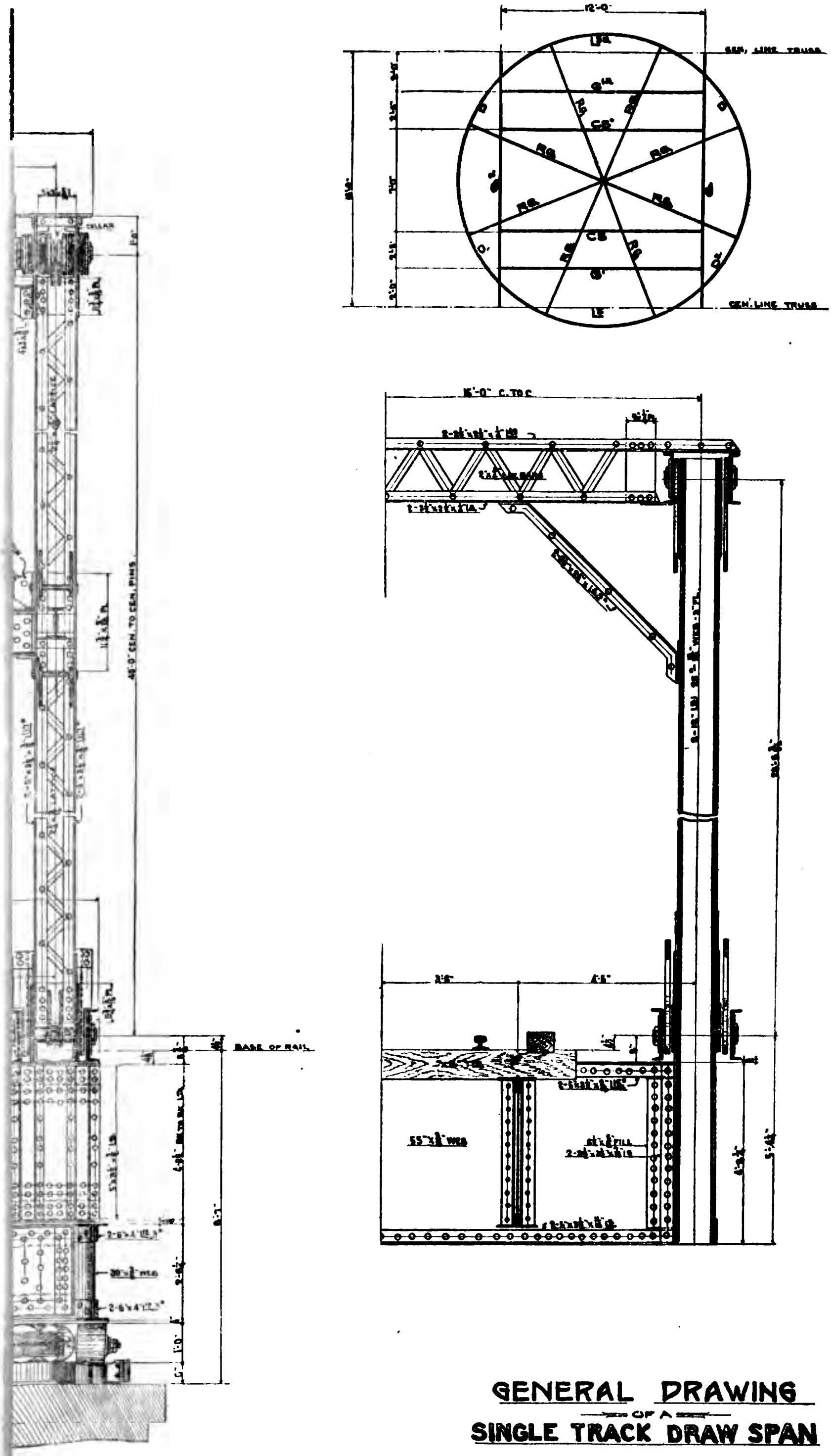


PLATE C.

290' SPAN.
SOUTHERN PACIFIC R. R.



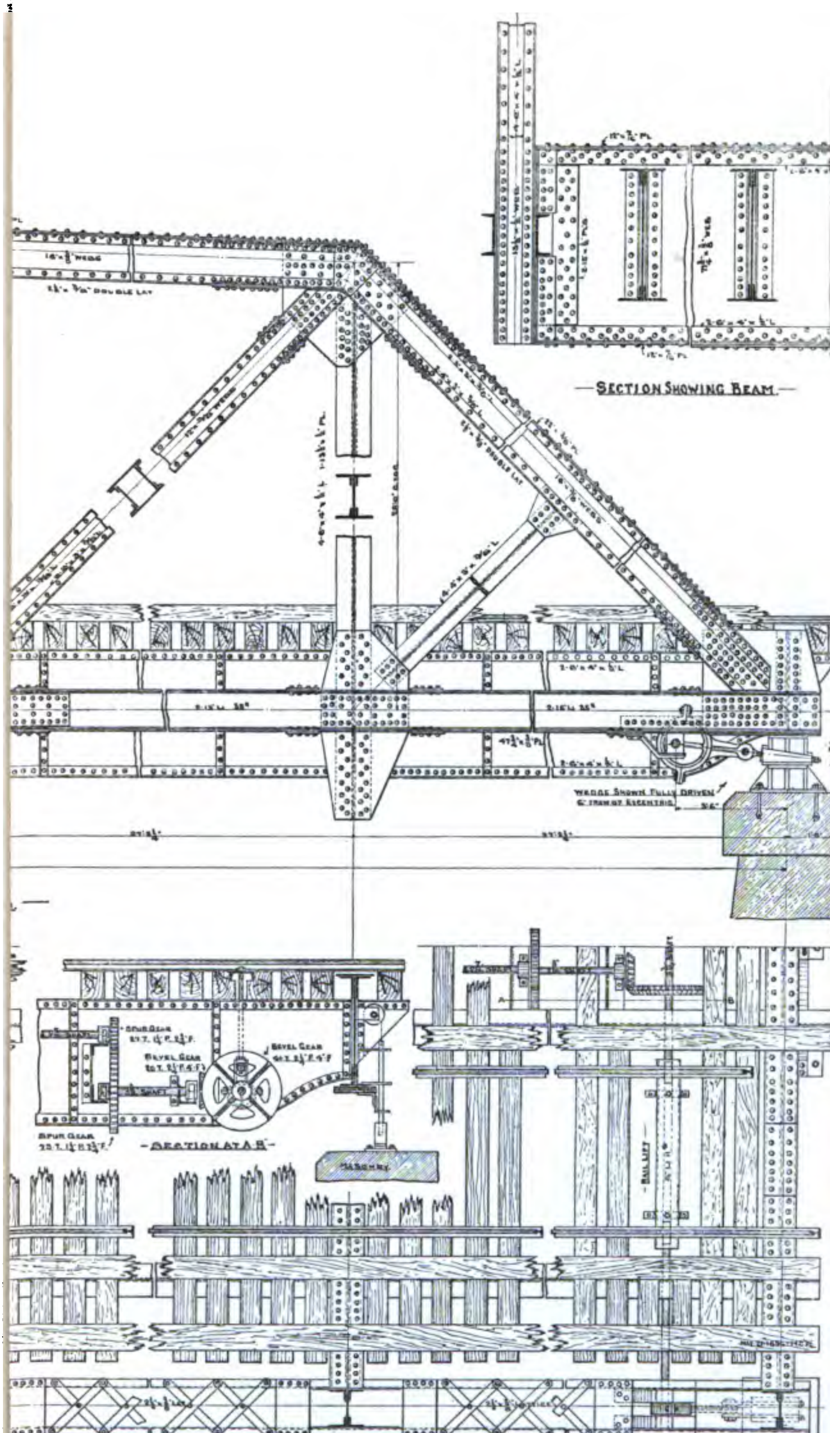
GENERAL DRAWING
OF A
SINGLE TRACK DRAW SPAN

290'-0½" C. TO C. END PINS – 16'0" WIDE.
40'0" DEEP AT CEN.

N OF CENTER.

PLATE D.

SOUTHERN PACIFIC R. R.



CHINERY — PLATE G.

DOUBLE-TRACK DRAW-SPAN OVER CALUMET RIVER.

EDGE MOOR BRIDGE WORKS, BUILDERS.

W. L. Stebbings, Con. Engineer.

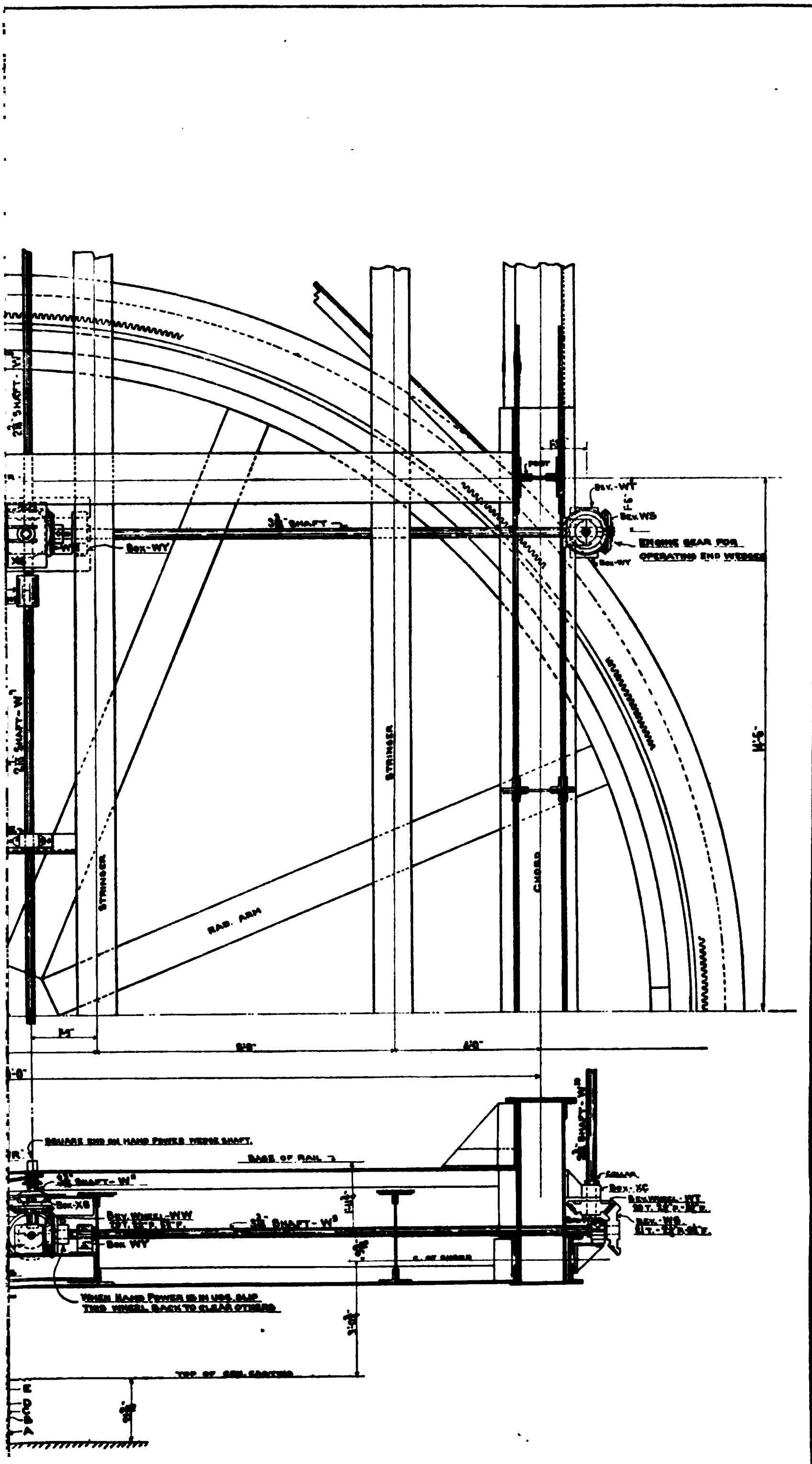
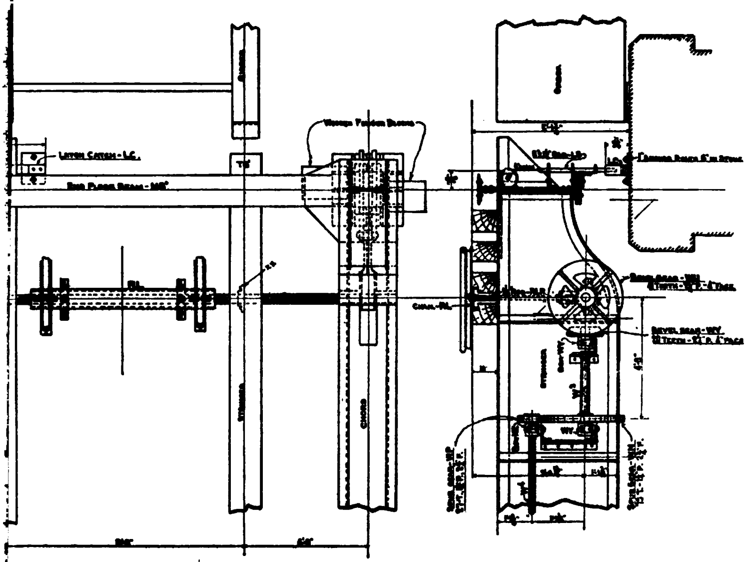


PLATE H.

CALUMET RIVER DRAW, CHICAGO.

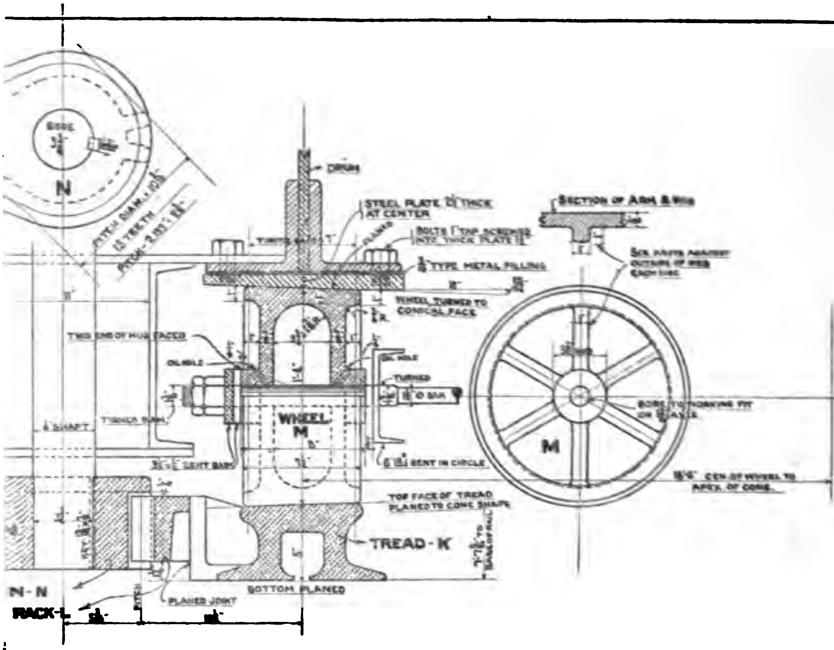
CALUMET AND BLUE ISLAND R. R.



RIVER DRAW, CHICAGO.

Latch and Wedging Machinery.

AND BLUE ISLAND R. R



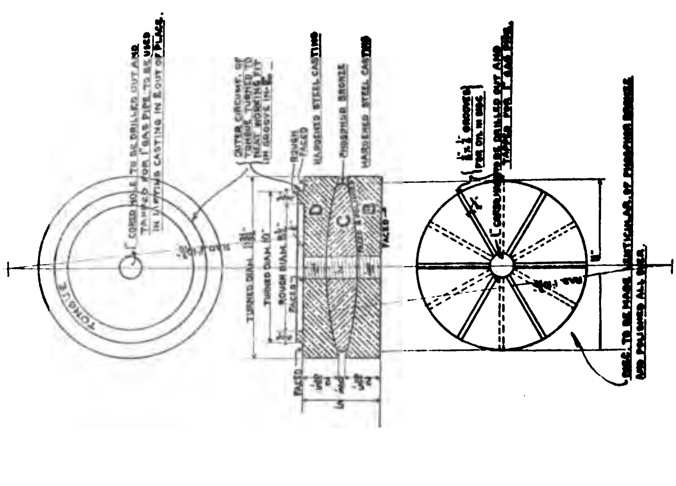


PLATE J.

CALUMET RIVER DRAW, CHICAGO.
CALUMET AND BLUE ISLAND R. R.

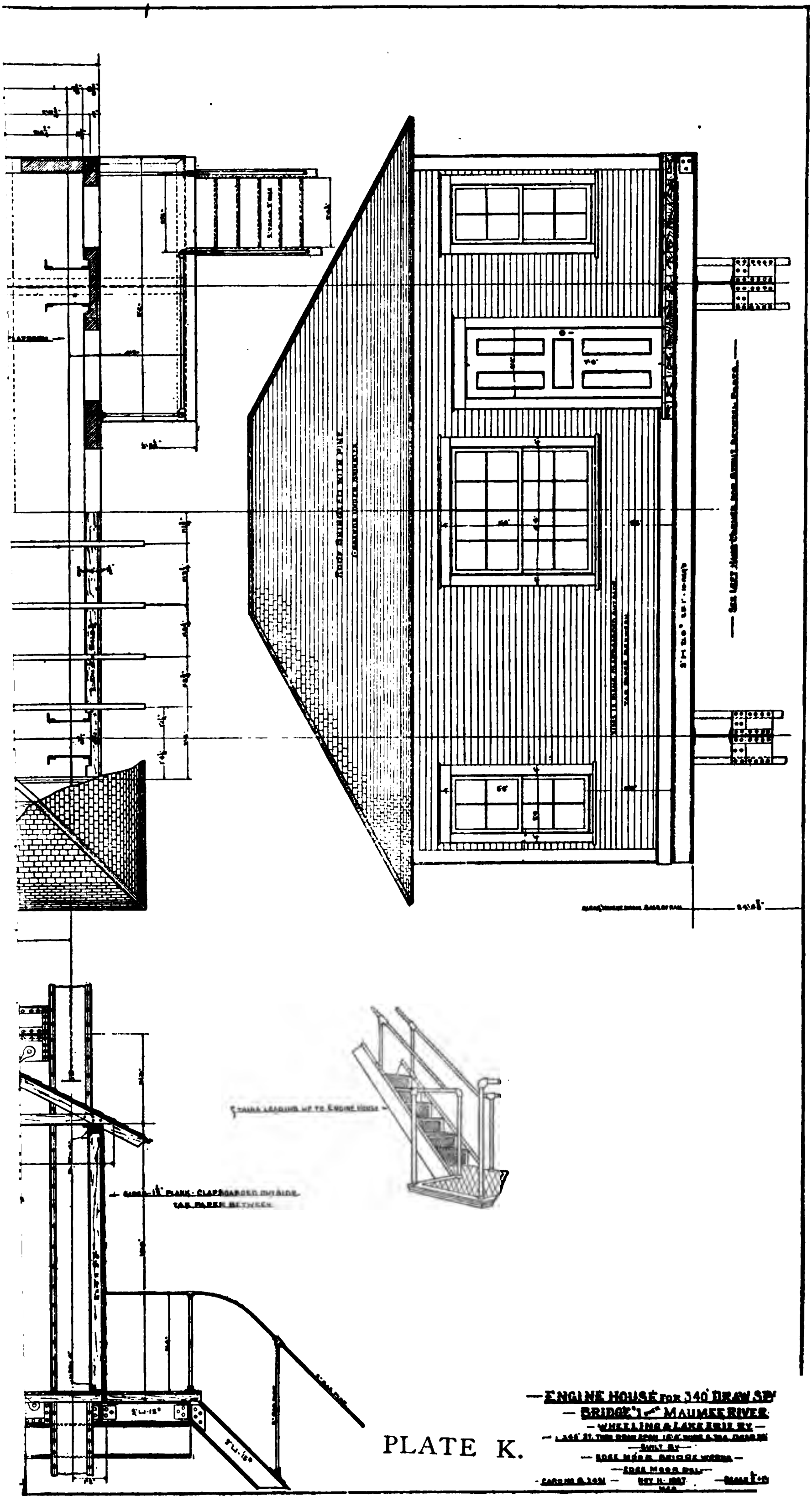
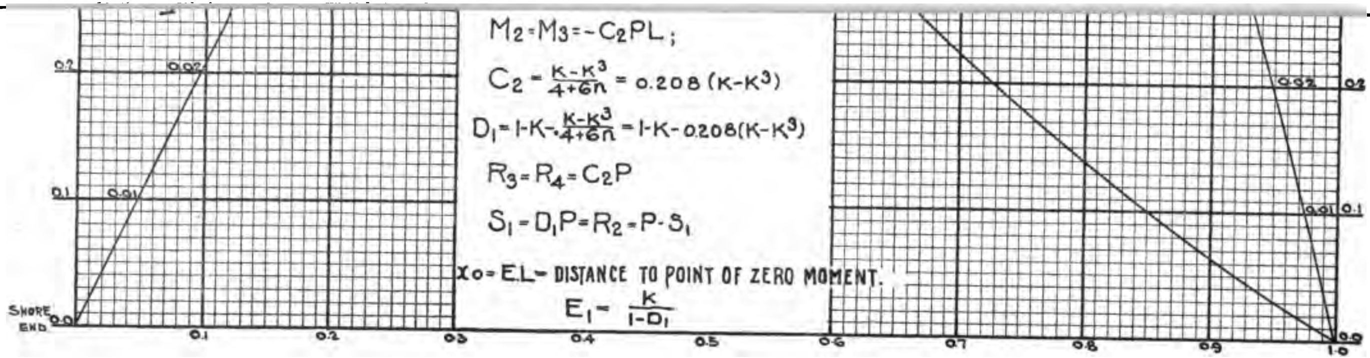
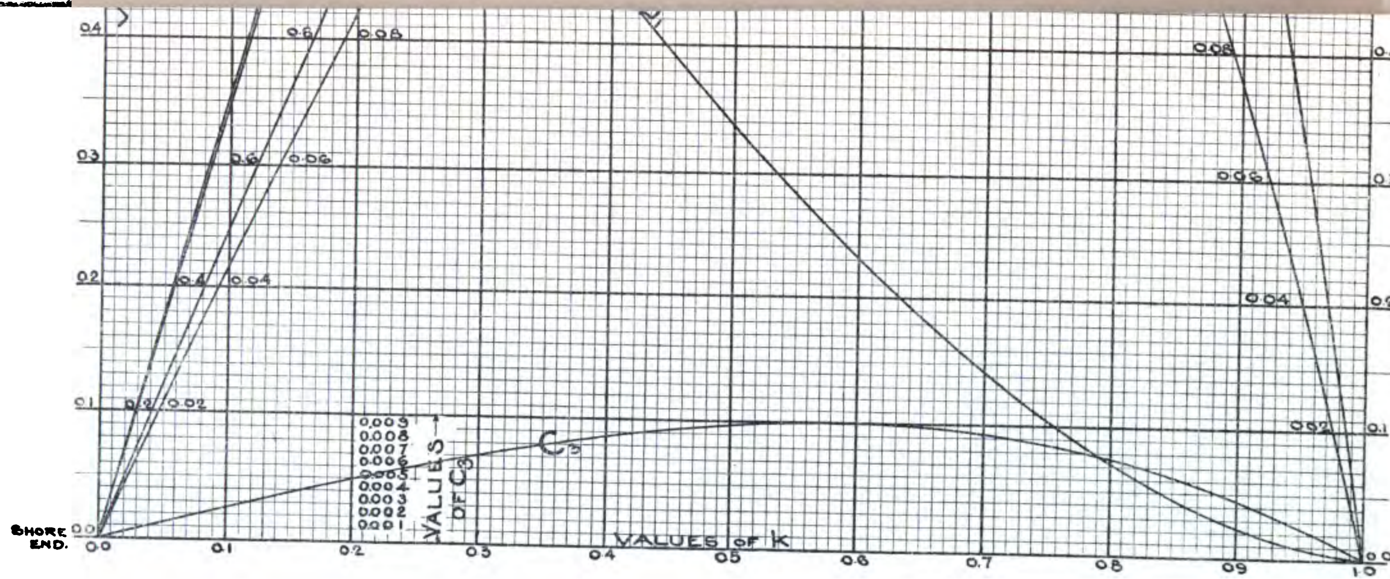
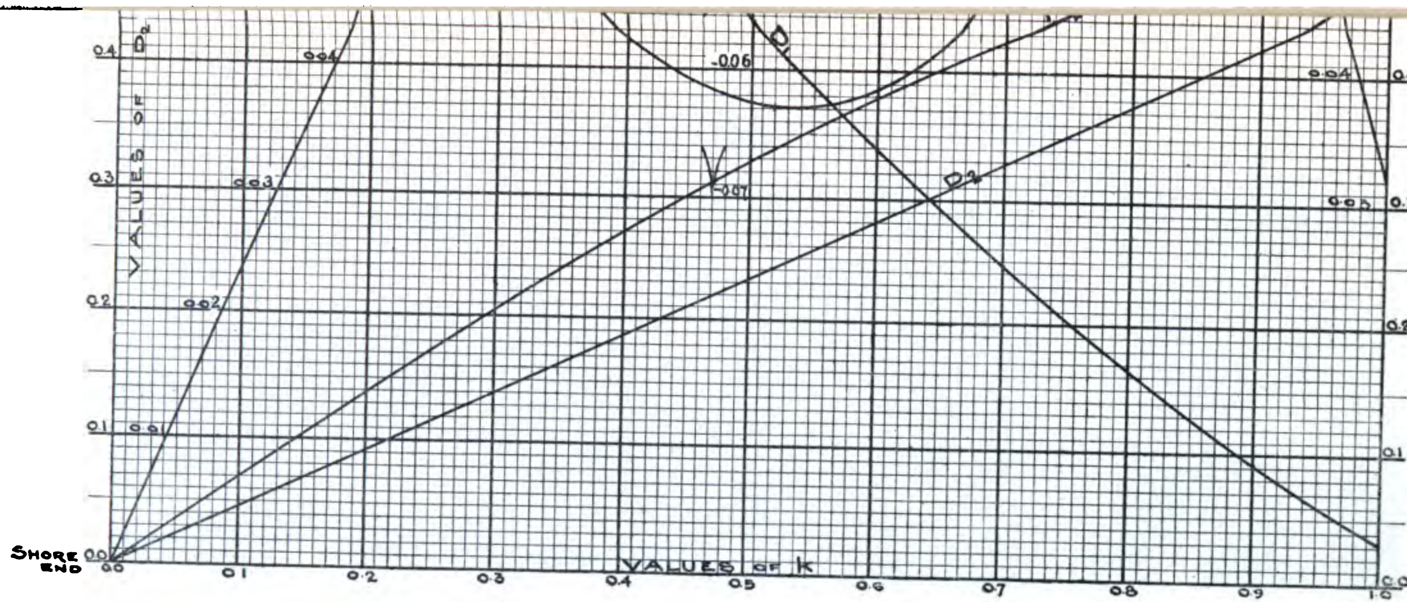

ENGINE HOUSE FOR 340' DRAW SP.
BRIDGE No 1 — MAUMEE RIVER
WHEELING & LAKE ERIE RY
BUILT BY
EDGE MOOR BRIDGE WORKS
EDGE MOOR DEL.

PLATE K.

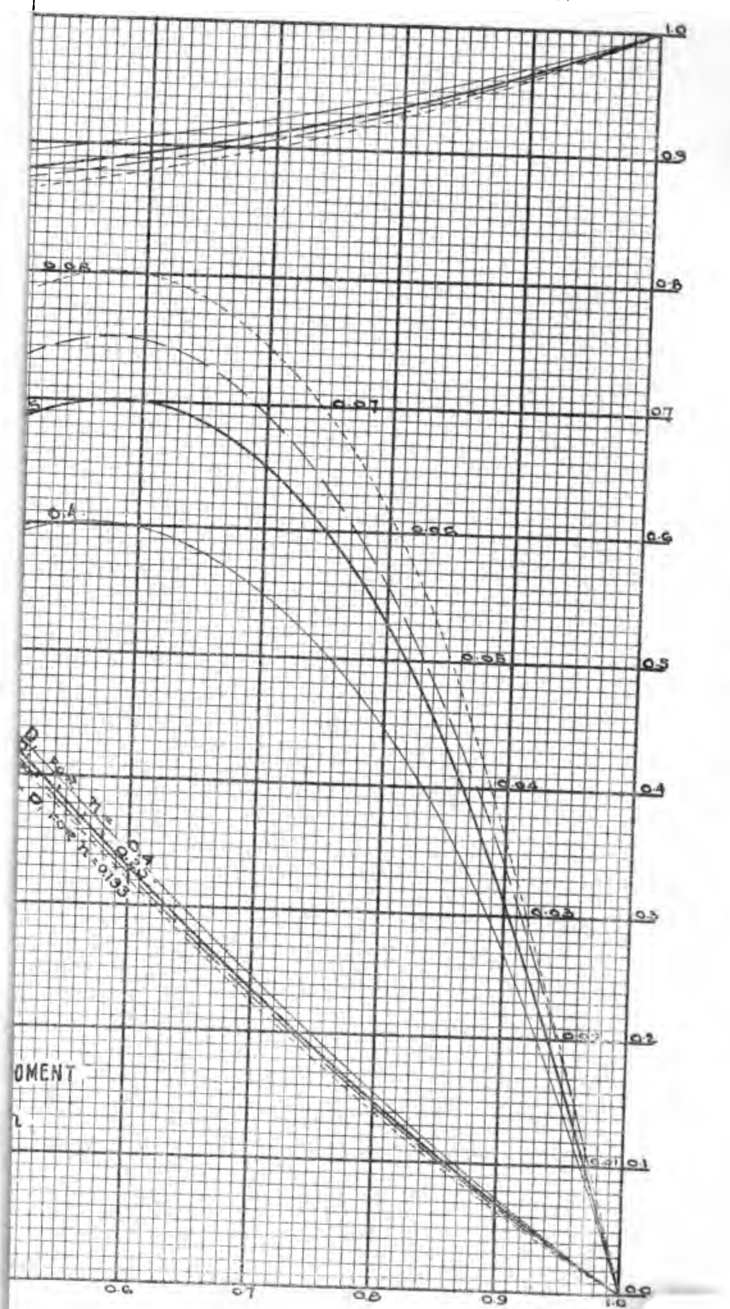

$M_2 = M_3 = -C_2PL;$
$C_2 = \frac{K-K^3}{4+6n} = 0.208\,(K-K^3)$
$D_1 = 1-K-\frac{K-K^3}{4+6n} = 1-K-0.208(K-K^3)$
$R_3 = R_4 = C_2P$
$S_1 = D_1P = R_2 = P-S_1$
$X_0 = E.L$ = DISTANCE TO POINT OF ZERO MOMENT.
$E_1 = \frac{K}{1-D_1}$
SHORE END
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
0.01
0.02



SHORE END.
VALUES OF K
VALUES OF C_3
C_3



VALUES OF D
VALUES OF K
SHORE END
D1
D2
V
-0.06
-0.07
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0



1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
0.08
0.07
0.06
0.05
0.04
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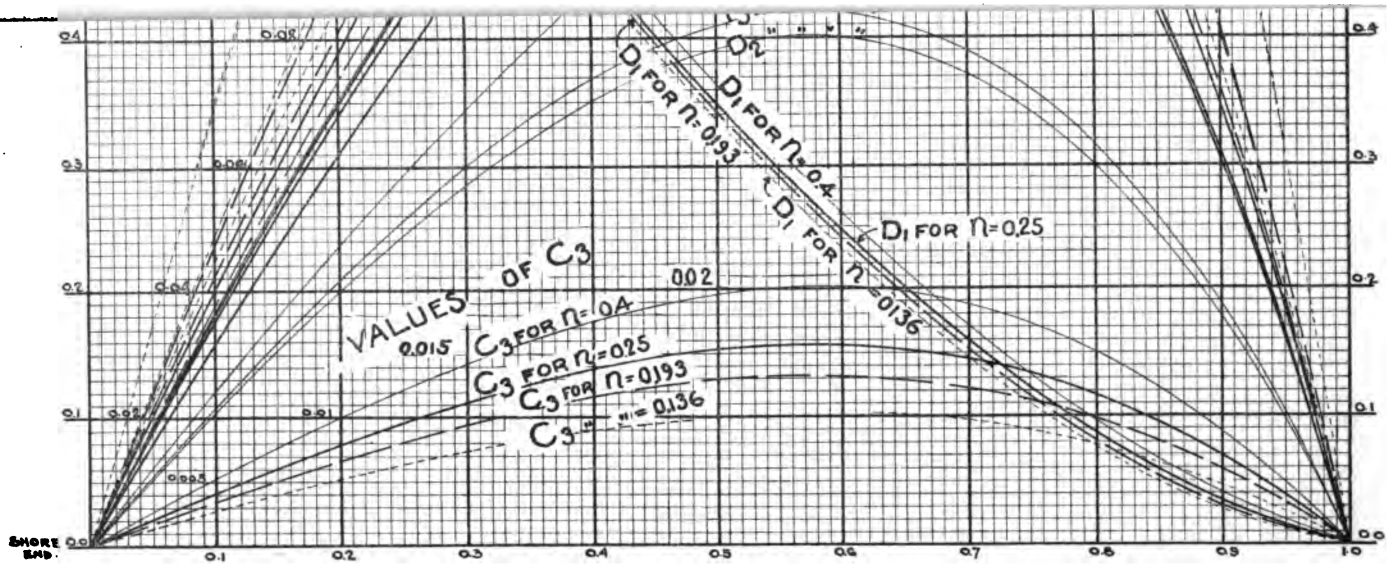
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